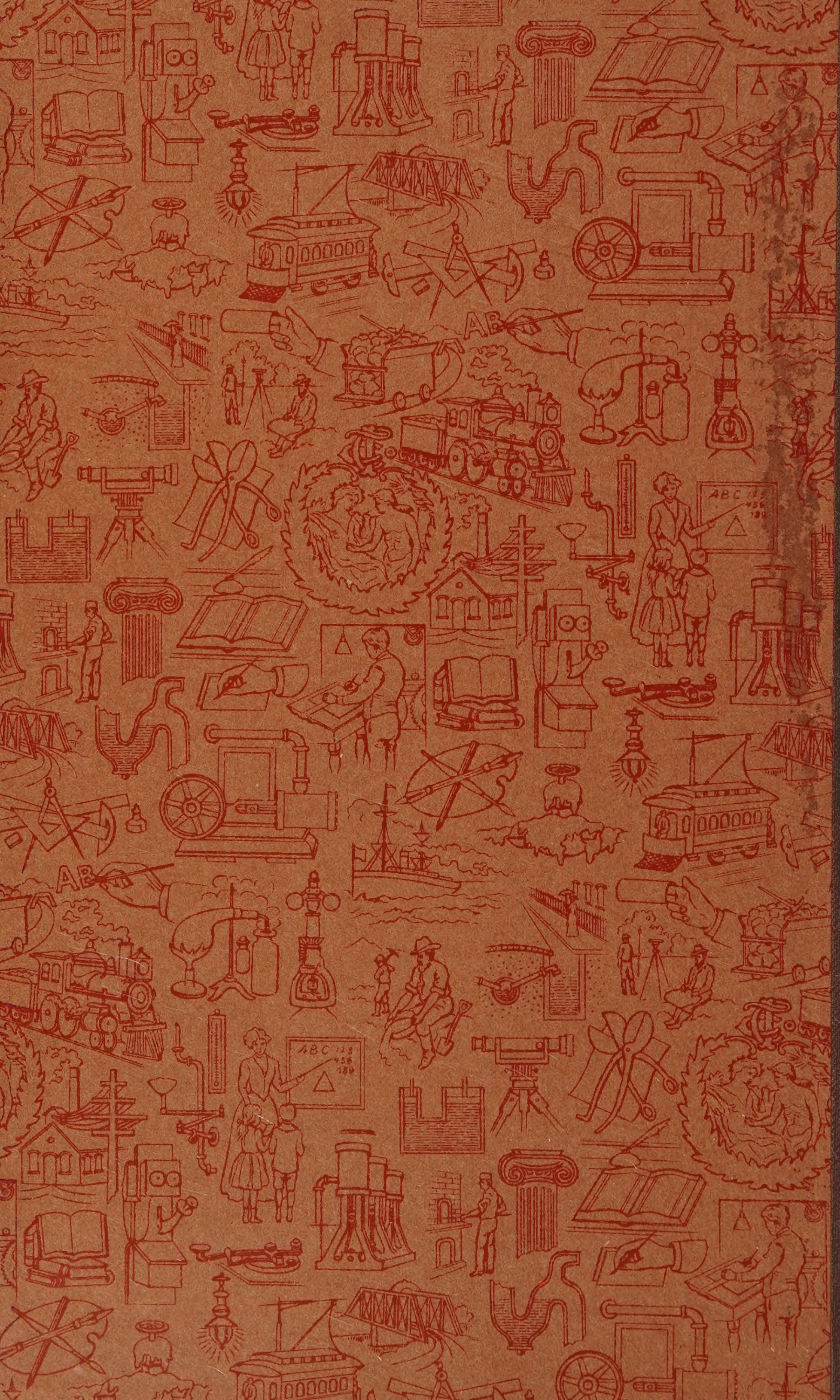
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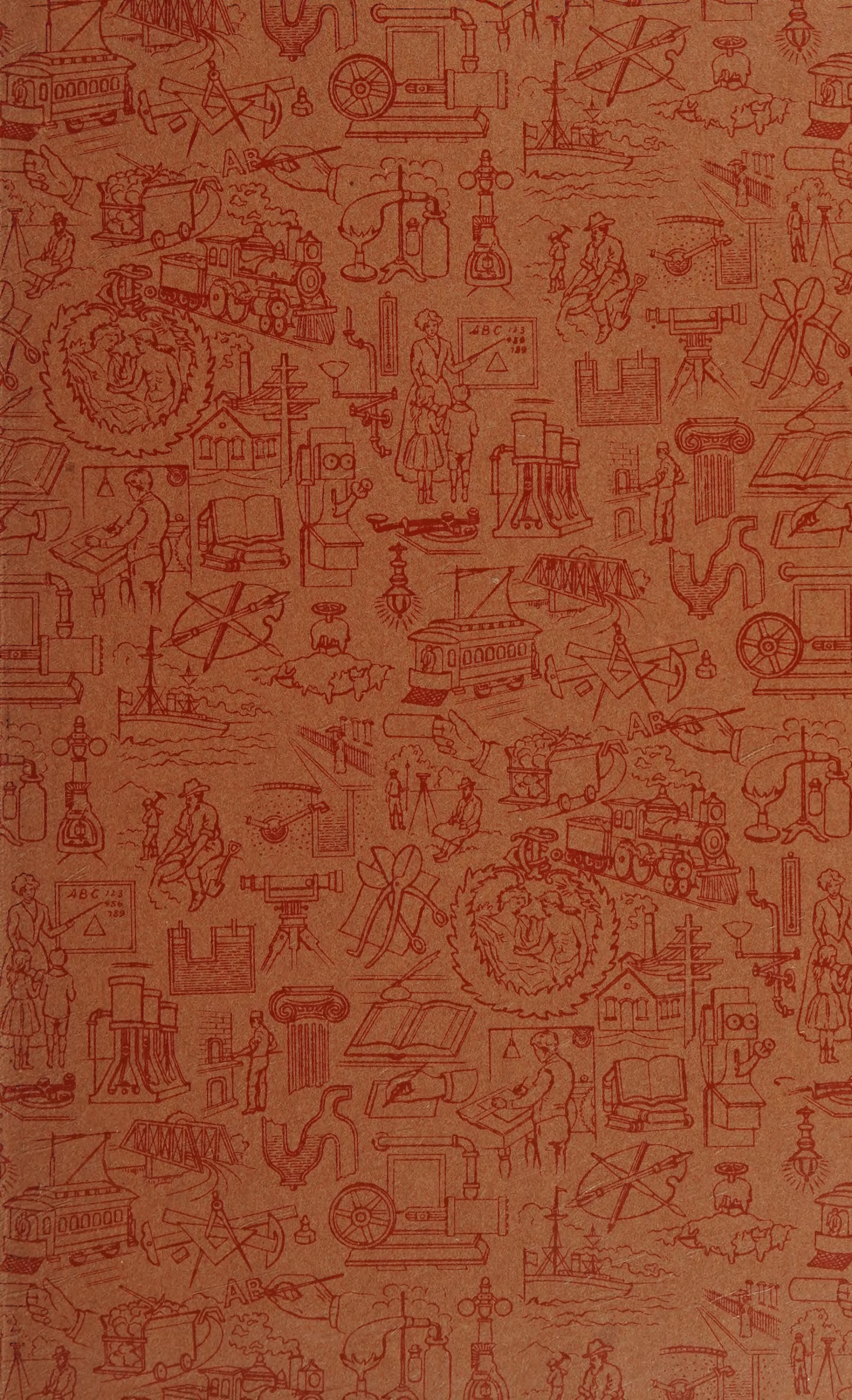
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PRINCIPLES OF MECHANICS.

MATTER AND ITS PROPERTIES.

DEFINITION OF MECHANICS.

1. Mechanics is that science which treats of the action of forces on bodies and the effects that they produce; it treats of the laws that govern the movement and equilibrium of bodies and shows how they may be applied.

MATTER.

2. Matter is anything that occupies space. It is the substance of which all bodies consist. Matter is composed of *molecules* and *atoms.*

3. A **molecule** is the smallest portion of matter that can exist without changing its nature.

4. An **atom** is an indivisible portion of matter.

Atoms unite to form molecules, and a collection of molecules forms a mass or body.

A drop of water may be divided and subdivided until each particle is so small that it can only be seen by the most powerful microscope, but each particle will still be water.

Now, imagine the division to be carried on still further, until a limit is reached beyond which it is impossible to go without changing the nature of the particle. The particle of water is now so small that, if it be divided again, it will

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cease to be water, and will be something else; we call this particle a *molecule.*

If a molecule of water be divided, it will yield 2 atoms of hydrogen gas and 1 of oxygen gas. If a molecule of sulphuric acid be divided, it will yield 2 atoms of hydrogen, 1 of sulphur, and 4 of oxygen.

5. Bodies are composed of collections of molecules. Matter exists in three conditions or forms: *solid*, *liquid*, and *gaseous.*

6. A **solid body** is one whose molecules change their relative positions with great difficulty; as iron, wood, stone, etc.

7. A **liquid body** is one whose molecules tend to change their relative positions easily. Liquids readily adapt themselves to the shape of vessels that contain them, and their upper surface always tends to become perfectly level. Water, mercury, molasses, etc., are liquids.

8. A **gaseous body,** or gas, is one whose molecules tend to separate from one another; as air, oxygen, hydrogen, etc.

Gaseous bodies are sometimes called **aeriform** (air-like) **bodies.** They are divided into two classes: the so-called "*permanent*" *gases* and *vapors.*

9. A **permanent gas** is one that remains a gas at ordinary temperatures and pressures.

10. A **vapor** is a body that at ordinary temperatures is a liquid or solid, but when heat is applied, becomes a gas, as steam.

One body may, under different conditions, exist in all three states; as, for example, mercury, which at ordinary temperatures is a liquid, becomes a solid (freezes) at 40° below zero, and a vapor (gas) at 600° above zero. By means of great cold, all gases, even hydrogen, have been liquefied, and many solidified.

By means of heat, all solids have been liquefied, and a great many vaporized. It is probable that, if we had the means of producing sufficiently great extremes of heat and cold, all solids might be converted into gases, and all gases into solids.

11. Every portion of matter possesses certain qualities called *properties*. Properties of matter are divided into two classes: *general* and *special*.

12. General properties of matter are those that are common to all bodies. They are as follows: *Extension*, *impenetrability*, *weight*, *indestructibility*, *inertia*, *mobility*, *divisibility*, *porosity*, *compressibility*, *expansibility*, and *elasticity*.

13. Extension is the property of occupying space. Since all bodies must occupy space, it follows that extension is a general property.

14. By **impenetrability** we mean that no two bodies can occupy exactly the same space at the same time.

15. Weight is the measure of the earth's attraction upon a body. All bodies have weight. In former times it was supposed that gases had no weight, since, if unconfined, they tend to move away from the earth, but, nevertheless, they will finally reach a point beyond which they cannot go, being held in suspension by the earth's attraction. Weight is measured by comparison with a standard. The standard is a bar of platinum weighing 1 pound, owned and kept by the Government.

16. Inertia means that a body cannot put itself in motion nor bring itself to rest. To do either it must be acted upon by some force.

17. Mobility means that a body can be changed in position by some force acting upon it.

18. Divisibility is that property of matter by virtue of which a body may be separated into parts.

19. Porosity is the term used to denote the fact that there is space between the molecules of a body. The molecules of a body are supposed to be spherical, and, hence, there is space between them, as there would be between peaches in a basket. The molecules of water are larger than those of salt; so that when salt is dissolved in water, its molecules wedge themselves between the molecules of the water, and, unless too much salt is added, the water will occupy no more space than it did before. This does not prove that water is penetrable, for the molecules of salt occupy the space that the molecules of water did not.

Water has been forced through iron by pressure, thus proving that iron is porous.

20. Compressibility is a natural consequence of porosity. Since there is space between the molecules, it is evident that by means of force (pressure) they can be brought closer together, and thus the body be made to occupy a smaller space.

21. Expansibility is the term used to denote the fact that the molecules of a body will, under certain conditions (when heated, for example), move farther apart, and so cause the body to *expand*, or occupy a greater space.

22. Elasticity is that property of matter which enables a body when distorted within certain limits to resume its original form when the distorting force is removed. Glass, ivory, and steel are very elastic, clay and putty in their natural state being very slightly so.

23. Indestructibility is the term used to denote the fact that we cannot *destroy* matter. A body may undergo thousands of changes, be resolved into its molecules, and its molecules into atoms, which may unite with other atoms to form other molecules and bodies entirely different in appearance and properties from the original body, but the same number of atoms remain. The whole number of atoms in the universe is exactly the same now as it was millions of years ago, and will always be the same. *Matter is indestructible.*

24. Special properties are those that are not possessed by all bodies. Some of the most important are as follows: *hardness*, *tenacity*, *brittleness*, *malleability*, and *ductility*.

25. Hardness.—A piece of copper will scratch a piece of wood, steel will scratch copper, and tempered steel will scratch steel in its ordinary state. We express all this by saying that steel is *harder* than copper, and so on. Emery and corundum are extremely hard, and the diamond is the hardest of all known substances. It can only be polished with its own powder.

26. Tenacity is the term applied to the power with which some bodies resist a force tending to pull them apart. Steel is very tenacious.

27. Brittleness. — Some bodies possess considerable power to resist either a pull or a pressure, but they are easily broken when subjected to shocks or jars; for example, good glass will bear a *greater* compressive force than most woods, but may be easily broken when dropped upon a hard floor; this property is called *brittleness*.

28. Malleability is that property which permits of some bodies being hammered or rolled into sheets. Gold is the most malleable of all substances.

29. Ductility is that property which enables some bodies to be drawn into wire. Platinum is the most ductile of all substances.

MOTION AND VELOCITY.

DEFINITIONS.

30. Motion is the opposite of rest and indicates a changing of position in relation to some object which is for that purpose regarded as being fixed. If a large stone is rolled down hill, it is in motion in relation to the hill.

If a person is on a railway train and walks in the opposite direction from that in which the train is moving, and with the same speed, he will be in motion as regards the train, but at rest with respect to the earth, since, until he gets to the end of the train, he will be directly over the spot at which he was when he started to walk.

31. The **path of a body in motion** is the line described by a certain point in the body called its center of gravity. No matter how irregular the shape of the body may be, nor how many turns and twists it may make, the line that indicates the direction of this point for every instant that it is in motion is the path of the body.

32. **Velocity** is rate of motion. It is measured by a unit of space passed over in a unit of time. When equal spaces are passed over in equal times, the velocity is said to be **uniform.**

If the flywheel of an engine keeps up a constant speed of a certain number of revolutions per minute, the velocity of any point is uniform. A railway train having a constant speed of 40 miles per hour moves 40 miles every hour, or $\frac{40}{60} = \frac{2}{3}$ mile every minute, and since equal spaces are passed over in equal times, the velocity is uniform.

33. **Variable Velocity.**—When a body moves in such a way that the spaces passed over in equal periods of time are unequal, its velocity is said to be **variable.**

34. The rate of motion of a body with a variable velocity may increase or decrease at a uniform rate. When the velocity varies in either of these ways, the body is said to have a **uniformly varying velocity.**

The most familiar example of uniformly varying velocity is a falling weight. Suppose a stone is dropped from a high bridge. It starts from a state of rest with no velocity, but its velocity constantly increases until it strikes. Its increase in velocity during any equal units of time is nearly the same. Thus, at the end of the first second its velocity will have increased from 0 to a rate of 32.16 feet per second, nearly; during the next second the velocity will have increased by the same amount, making the velocity at the end of the second second 64.32 feet. At the end of the third second a like increase will have taken place, and the velocity will then be 3 × 32.16 = 96.48 feet per second.

35. The change in the velocity of a body during a period of time is called its **acceleration** for that period. Thus, in the case of the falling weight just considered, the change of 32.16 feet per second that takes place in its velocity during each second is its acceleration in feet per second for each second considered. If the period of time considered had been 2 seconds, the acceleration would have been the increase in velocity during this time, that is, 64.32 feet per second for each 2 seconds considered.

36. The change in velocity may be from a higher to a lower rate. Thus, when a stone is thrown upwards, it leaves the hand with a given velocity; its upward motion is constantly resisted by the force of gravity and the resistance of the air, and in consequence of these resistances, it moves slower and slower until it finally stops and begins to return to the earth. A change in velocity of this kind is sometimes called **retardation.**

37. The **mean** or **average velocity** of a body moving with a variable velocity can only be given for a stated period of time, and is numerically equal to the uniform velocity that will take the body over the same distance in the same time.

RULES FOR VELOCITY PROBLEMS.

38. Uniform and Average Velocity.—

Let $d =$ distance;
$t =$ time;
$v =$ velocity.

Rule 1.—*To find the uniform or the average velocity that a body must have to pass over a certain distance or space in a given time, divide the distance by the time.*

Or, $$v = \frac{d}{t}.$$

Example 1.—The piston of a steam engine travels 3,000 feet in 5 minutes; what is its average velocity in feet per minute?

Solution.—Here 3,000 feet is the distance, and 5 minutes is the time. Applying the rule, $3,000 \div 5 = 600$ ft. per min. Ans.

Caution.—Before applying the above or any of the succeeding rules, care must be taken to reduce the values given to the denominations required in the answer. Thus, in the above example, if the velocity is required in feet per second instead of in feet per minute, the 5 minutes must be reduced to seconds before dividing. The operation will then be: 5 minutes $= 5 \times 60 = 300$ seconds. Applying the rule, $3,000 \div 300 = 10$ ft. per sec. Ans.

If the velocity is required in inches per second, it is necessary to reduce the 3,000 feet to inches and the 5 minutes to seconds, before dividing. Thus, 3,000 feet $\times 12 = 36,000$ inches. 5 minutes $\times 60 = 300$ seconds. Now applying the rule, $36,000 \div 300 = 120$ in. per sec. Ans.

Example 2.—A railroad train travels 50 miles in 1½ hours; what is its average velocity in feet per second?

Solution.—Reducing the miles to feet and the hours to seconds, 50 miles $\times 5,280 = 264,000$ feet. 1½ hours $\times 60 \times 60 = 5,400$ seconds. Applying the rule, $264,000 \div 5,400 = 48\frac{8}{9}$ ft. per sec. Ans.

39. If the uniform velocity (or the average velocity) and the time are given, and it is required to find the distance that a body having the given velocity will travel in the given time:

Rule 2.—*Multiply the velocity by the time.*

Or,
$$d = vt.$$

Example 1.—The velocity of sound in still air is 1,092 feet per second; how many miles will it travel in 16 seconds?

Solution.—Reducing the 1,092 feet to miles, $1,092 \div 5,280 = \frac{1092}{5280}$. Applying the rule, $\frac{1092}{5280} \times 16 = 3.31$ mi., nearly. Ans.

Example 2.—The piston speed of an engine is 11 feet per second, how many miles does the piston travel in 1 hour and 15 minutes?

Solution.— 1 hour and 15 minutes reduced to seconds $= 4,500$ seconds = the time. 11 feet reduced to miles $= \frac{11}{5280}$ mile = velocity in miles per second. Applying the rule, $\frac{11}{5280} \times 4,500 = 9.375$ mi. Ans.

40. If the distance through which a body moves is given, and also its average or uniform velocity, and it is desired to know how long it takes the body to move through the given distance:

Rule 3.—*Divide the distance, or space passed over, by the velocity.*

Or, $$t = \frac{d}{v}.$$

EXAMPLE 1.—Suppose that the radius of the crank of a steam engine is 15 inches and that the shaft makes 120 revolutions per minute; how long will it take the crankpin to travel 18,849.6 feet?

SOLUTION.—Since the radius, or distance from the center of the shaft to the center of the crankpin, is 15 inches, the diameter of the circle it moves in is 15 inches × 2 = 30 inches = 2.5 feet. The circumference of this circle is 2.5 × 3.1416 = 7.854 feet. 7.854 × 120 = 942.48 feet = distance that the crankpin travels in 1 minute = velocity in feet per minute. Applying the rule, 18,849.6 ÷ 942.48 = 20 min. Ans.

EXAMPLE 2.—A point on the rim of an engine flywheel travels at the rate of 150 feet per second; how long will it take it to travel 45,000 feet?

SOLUTION.—Applying the rule,

45,000 ÷ 150 = 300 sec. = 5 min. Ans.

EXAMPLES FOR PRACTICE.

1. A locomotive has drivers 80 inches in diameter. If they make 293 revolutions per minute, what is the velocity of the train in (*a*) feet per second? (*b*) miles per hour?

$$\text{Ans.}\begin{cases}(a) & 102.277 \text{ ft. per sec.}\\ (b) & 69.734 \text{ mi. per hr.}\end{cases}$$

2. Assuming the velocity of steam as it enters the cylinder to be 900 feet per second, how far can it travel, if unobstructed, during the time the flywheel of an engine revolves 7 times, if the number of revolutions per minute are 120? Ans. 3,150 ft.

3. The average speed of the piston of an engine being 528 feet per minute, how long will it take the piston to travel 4 miles? Ans. 40 min.

4. A speed of 40 miles per hour equals how many feet per second? Ans. 58⅔ ft.

5. The earth turns around once in 24 hours. If the diameter be taken as 8,000 miles, what is the velocity of a point on the equator in miles per minute? Ans. 17.45⅓ mi. per min.

6. The stroke of an engine is 28 inches. If the engine makes 11,400 strokes per hour, (*a*) what is its speed in feet per minute? (*b*) how far will this piston travel in 11 minutes?

$$\text{Ans.}\begin{cases}(a) & 443\tfrac{1}{3} \text{ ft. per min.}\\ (b) & 4{,}876 \text{ ft. } 8 \text{ in.}\end{cases}$$

FORCE.

GENERAL PRINCIPLES.

41. Force is known only by the effects it produces on matter. Forces cannot, therefore, be compared in the same way that quantities of matter or distances are compared; the only way in which forces can be examined in reference to one another is by noting their relative effects in the production of motion or a change of state in matter.

The most familiar conception of force is that of a push or pull that tends to produce or destroy motion. If the force is great enough, its effect is seen in a change in the state of motion of the body on which it acts; that is, it either produces motion in the body or destroys some or all of the motion already existing. If, however, the body acted on by a force is so situated that the force applied to it is opposed by a resisting force of equal magnitude, no change in motion is produced. In this case the force is not perceivable, unless some other force is introduced whose effects will reveal the existence of the first force.

42. Forces are called by various names according to the ways in which they manifest themselves. Manifestations of force are: *attraction*, *repulsion*, *cohesion*, *adhesion*, *acceleration*, *retardation*, *resistance*, etc., and the forces producing these manifestations are called *attractive*, *repulsive*, *cohesive*, *adhesive*, *accelerating*, *retarding*, *resisting*, etc. forces.

43. Comparison of Forces.—In considering the effects of a force on a body, some standard of comparison must be used. The standard most commonly adopted in English-speaking countries is the **pound**, which is the force required to raise a standard mass of matter from the ground under certain specified conditions.

In practice, force is always regarded as a pressure; that is, a force is considered the equivalent of the pressure exerted by a weight. For example, the effect of a force of

20 pounds acting upon a body is the same as the pressure of 20 pounds exerted by a weight of 20 pounds.

44. In order that we may compare the effect of a force on a body with that of another force on another body, it is necessary that the following *three conditions* be fulfilled in regard to both bodies:

1. The point of application, or point at which the force acts upon the body, must be known.
2. The direction of the force, or, what is the same thing, the straight line along which the force tends to move the point of application, must be known.
3. The magnitude or value of the force in comparison with the given standard must be known.

45. Reduction of Forces.—When a number of forces act upon a body and produce a certain effect, it is often necessary to find a *single* force that, when substituted for the given forces, will produce the *same* effect. In order to find the point of application, the direction, and the magnitude of this single force, it is necessary to know the above three conditions of every one of the given forces.

NEWTON'S LAWS OF MOTION.

46. The fundamental principles of the relations between force and motion were first stated by Sir Isaac Newton, and are called *Newton's Three Laws of Motion.* They are as follows:

1. *All bodies continue in a state of rest, or of uniform motion in a straight line, unless acted upon by some external force that compels a change.*
2. *Every motion or change of motion is proportional to the acting force, and takes place in the direction of the straight line along which the force acts.*
3. *To every action there is always opposed an equal reaction.*

These laws in their accepted form, as just given, have been more or less indirectly derived from experiment, but they are so comprehensive as to defy complete experimental verification.

47. Exemplification of the First Law.—In the first law of motion, it is stated that a body once set in motion by any force, no matter how small, will move forever in a straight line, and always with the same velocity, unless acted upon by some other force that compels a change. It is not possible to actually verify this law, on account of the earth's attraction for all bodies, but, from astronomical observations, we are certain that the law is true. This law is often called the **law of inertia.**

48. The word **inertia** is so abused that a full understanding of its meaning is important. Inertia is not a force, although it is often so called. If a force acts upon a body and puts it in motion, the effect of the force is stored in the body, and a second body, in stopping the first, will receive a blow equal in every respect to the original force, assuming that there has been no resistance of any kind to the motion of the first body.

It is dangerous for a person to jump from a fast-moving train, for the reason that, since his body has the same velocity as the train, it has the same force stored in it that would cause a body of the same weight to take the same velocity as the train, and the effect of a sudden stoppage is the same as the effect of a blow necessary to give the person that velocity.

By "bracing" himself and jumping in the same direction that the train is moving, and running, he brings himself gradually to rest, and thus reduces the danger. If a body is at rest, it must be acted upon by a force in order to be put in motion, and, no matter how great the force may be, it cannot be *instantly* put in motion.

The resistance thus offered to being put in motion is commonly, but erroneously, called the *resistance of inertia.* It should be called the *resistance due to inertia.*

49. Exemplification of the Second Law.—From the second law, we see that if two or more forces act upon a body, their final effect on the body will be in proportion to their magnitudes and to the directions in which they act.

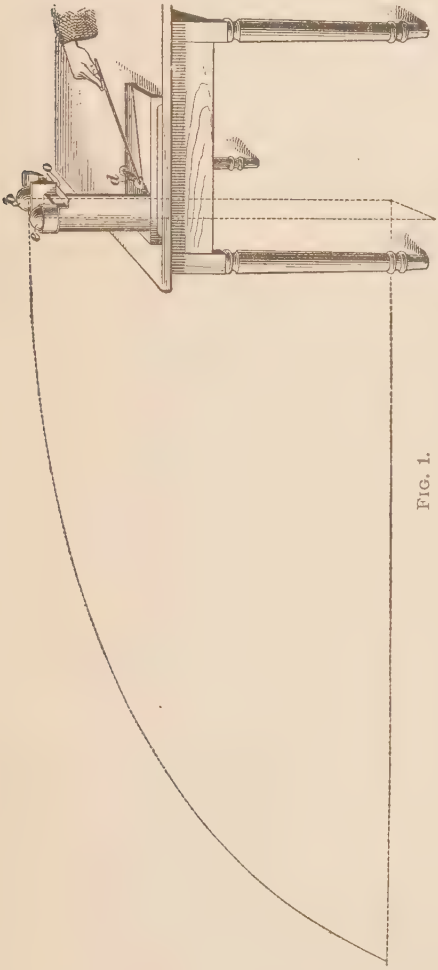


FIG. 1.

Thus, if the wind is blowing due west with a velocity of 50 miles per hour, and a ball is thrown due north with the same velocity, or 50 miles per hour, the wind will carry the ball west while the force of the throw is carrying it north, and the combined effect will be to cause it to move northwest.

The amount of departure from due north will be proportional to the force of the wind and independent of the velocity due to the force of the throw.

50. In Fig. 1 a ball *e* is supported in a cup, the bottom of which is attached to the lever *o* in such a manner that *o* will swing the bottom horizontally and allow the ball to drop. Another ball *b* rests in a horizontal groove that is provided with a slit in the bottom. A swinging arm is actuated by the spring *d* in such a manner that, when drawn back, as shown, and then released, it will strike the lever *o* and the ball *b* at the same time. This gives *b* an impulse in a horizontal direction, and swings *o* so as to allow *e* to fall.

On trying the experiment, it is found that *b* follows a path shown by the curved dotted line, and reaches the floor at the same instant as *e*, which drops vertically. This shows that the force that gave the first ball its horizontal movement had no effect on the vertical force that compelled both balls to fall to the floor; the vertical force produces the same effect as if the horizontal force had not acted. The second law may also be stated as follows: *A force has the same effect in producing motion, whether it acts upon a body at rest or in motion, and whether it acts alone or with other forces.*

51. Exemplification of the Third Law.—The third law states that action and reaction are equal. A man cannot lift himself by his boot straps for the reason that he presses downwards with the same force that he pulls upwards; the downward reaction equals the upward action, and is opposite to it.

In springing from a boat, we must exercise caution or the reaction will drive the boat from the shore. When we jump from the ground, we tend to push the earth from us, while the earth reacts and pushes us from it.

EXAMPLE.—Two men pull on a rope in opposite directions, each exerting a force of 100 pounds; what is the force that the rope resists?

SOLUTION.—Imagine the rope to be fastened to a tree, and that one man pulls with a force of 100 pounds. The rope evidently resists 100 pounds. According to Newton's third law, the reaction of the tree is also 100 pounds. Now, suppose the rope to be slacked, but that one end is still fastened to the tree; the second man then takes hold of the rope near the tree, and pulls with a force of 100 pounds, the first man pulling as before. The resistance of the rope is 100 pounds, as before, since the second man merely takes the place of the tree. He is obliged to exert a force of 100 pounds to keep the rope from slipping through his fingers. If the rope is passed around the tree, and each man pulls an end with a force of 100 pounds in the same and parallel directions, the stress in the rope is 100 pounds, as before, but the tree must resist the pull of both men, or 200 pounds.

52. **Dynamics,** also called **kinetics,** is that branch of mechanics that treats of forces and their effects when they *produce a change in motion* in the bodies on which they act.

53. **Statics** is that branch of mechanics that treats of forces and their effects when they do *not produce a change in motion* in the bodies on which they act.

GRAVITATION AND WEIGHT.

54. Every body in the universe exerts a certain attractive force on every other body, which tends to draw the two together. To scientists, this attractive force is known as **gravitation.**

If a body is held in the hand, a downward pull is felt, and if the hold is loosened, the body will fall to the ground. This pull, which we commonly call **weight,** is the attraction between the earth and the body.

55. The attraction between the earth and bodies at or near its surface is denoted by the term **force of gravity.** This attraction is generally considered as acting along the line joining the center of gravity of the body and the center of the earth. By **center of gravity** is meant that point of a body at which its whole weight may be assumed to be concentrated.

56. The weight of a body is directly proportional to the force of gravity. From this it follows that the weight of a body can only be uniform everywhere if the force of gravity is uniform. As a matter of fact, the force of gravity varies in different locations; consequently, the weight of the body is not the same at all points on the surface of the earth. This has been conclusively shown by sensitive spring balances.

ACCELERATING AND RETARDING FORCES.

57. According to the first law of motion, if a body is set in motion by a force and the force then ceases to act, the body will continue to move at the rate it had at the instant the action of the force was discontinued, unless acted upon by some other force.

If, however, a force acts upon a body for a given period of time, say 1 second, and imparts to it a certain amount of motion, and then, instead of ceasing to act, acts with the same intensity during the next second, it will impart to the body an increase in velocity equal to the velocity imparted during the first second. Then, the velocity at the end of the second second will be twice that at the end of the first. During the third second a like increase in velocity will be produced, making the velocity at the end of the third second three times as great as at the end of the first. This uniform increase in velocity will continue as long as the constant force continues to act on the body. A constant force when producing a constant acceleration is called a **constant accelerating force.**

58. If a constant force is applied to a body in motion in such a manner that it opposes the motion, its effect will be to reduce the motion by a certain amount, which will be the same for each second during which it acts. In this case, the force is called a **constant retarding force.**

We thus see that the effect of a constant force acting upon a body in motion is to produce a uniform acceleration or retardation in the velocity of motion of the body, it being assumed that the motion of the body is not opposed by varying resisting forces.

59. Acceleration Due to the Force of Gravity.—If a body falls freely under the action of the force of gravity, its velocity will increase at a uniform rate; in other words, it will be accelerated. Since the force of gravity varies in different localities, it follows that the acceleration produced by it is not everywhere the same. The greatest range in the acceleration due to the force of gravity in the United States is from a minimum of about 32.089 feet per second up to a maximum of about 32.186 feet per second for each second. In the latitude of Scranton, Pa., and at the level of the sea, the acceleration is nearly 32.16 feet per second;

this value will be used in all calculations in this Course that involve the use of acceleration due to the force of gravity. In accordance with the practice of most scientific writers, we will denote the acceleration due to the force of gravity by the letter g.

60. Mass.—If the weight of a body at any place, as determined by a spring balance, is divided by the acceleration due to the force of gravity at that place, a numerical value will be obtained that, for the same body, will be the same wherever it may be weighed. This quotient is called the **mass** of the body, and is generally designated by the letter m.

Rule 4.—*To find the mass of a body, divide its weight by the acceleration due to the force of gravity.*

Let W = the weight of a body;
g = acceleration due to gravity;
m = mass of the body.

Then,
$$m = \frac{W}{g}.$$

EXAMPLE.—What is the mass of a body weighing 96.48 pounds?

SOLUTION.—Applying the rule, $m = \frac{96.48}{32.16} = 3$. Ans.

61. Law of Gravitation.—The attractive force by which one body tends to draw another body toward it is directly proportional to its mass and inversely proportional to the square of the distance between their centers of gravity.

62. Laws of Weight.—

1. *Bodies weigh most at the surface of the earth. Below the surface, the weight decreases directly as the distance to the center of the earth decreases.*
2. *Above the surface, the weight decreases inversely as the square of the distance.*

63. Change in Motion of a Body.—A change in the motion of a body cannot take place without the action of an accelerating or retarding force. The force required to produce a given acceleration or retardation in a body is given by the following rule, where

f = force in pounds;
a = acceleration or retardation in feet per second.

Rule 5.—*Multiply the mass of the body by the acceleration, or retardation, in feet per second.*

Or, $$f = m a.$$

Since $m = \frac{W}{g}$ (see rule 4), this may also be written $f = \frac{W}{g} a$.

EXAMPLE.—What force will be required to give a body weighing 90 pounds an acceleration of 5 feet per second?

SOLUTION.—Applying rule **5**, we get

$$f = \frac{90}{32.16} \times 5 = 13.99+ \text{ lb., say } 14 \text{ lb.} \quad \text{Ans.}$$

64. According to the first law of motion, a body in motion not acted upon by any external force will continue its motion without any further application of a force. In practice, however, the motion of a body is always opposed by some resisting force or forces. According to the third law of motion, the force required to overcome the resistance is equal to the resistance.

The opposing forces are usually constant, or nearly so. Taking the opposing forces into account, the actual force required to accelerate a body meeting with resistance will be the sum of the accelerating force and the opposing forces.

ILLUSTRATION:—Imagine a weight of 321.6 pounds to be lying on a smooth plane surface. Assume that it has been determined experimentally that a force of 100 pounds is required to be exerted continually to overcome the friction between the weight and the surface. What force will be required to produce an acceleration of 2 feet per second?

By rule **5**, the accelerating force is $\frac{321.6}{32.16} \times 2 = 20$ pounds. As a force of 100 pounds must be exerted continually to overcome the resistance due to friction, a force of $100 + 20 = 120$ pounds will be required to produce the required acceleration.

65. The question is often asked, what force is required to start a flywheel and keep it going at a stated number of revolutions per minute? This question, or similar questions, cannot be answered without knowing the time in which the flywheel is to attain the given speed. An accelerating force depending on the mass of the wheel, its diameter, the distribution of the material of which it is made, the time of acceleration, and the constant resistances, is required to bring the wheel up to its speed. When this speed has been attained, the force required to keep it going will be that required to overcome the frictional and air resistances.

MOMENTUM.

66. Experience teaches us that the same force acting upon bodies of different weights produces different effects. For example, if a given force imparts a velocity of 10 feet per second in a certain time to a body weighing 1 pound, we know from experience and observation that it cannot impart the same velocity in the same time when acting upon a body weighing 1,000 pounds.

Scientists have shown that the velocity imparted to a body in a given time by a force *varies directly as the force* and *inversely as the mass* of the body. Hence, forces may be compared with one another by comparing their effects in imparting velocities to bodies whose masses are known.

67. The product obtained by multiplying the mass of a body by its velocity in feet per second is called the **momentum** of the body; it represents the magnitude of the force that will produce the given velocity in the body in 1 second. Hence, we may call momentum the **time effect** of a force.

68. According to the third law of motion, action and reaction are equal to each other. Consequently, if the force required to produce a stated momentum in a given time is known, it is likewise known what force is required to *destroy* this momentum, or to bring the body to rest, in an equal period of time.

When a liquid body is flowing in a stream, as from a nozzle, the weight to be considered in problems involving momentum is the weight of the liquid discharged in 1 second. For example, let it be required to estimate the force with which a man must hold the nozzle of a fire hose to prevent its slipping through his hands when a stream of water issues from it with a velocity of 20 feet per second, the area of the opening in the nozzle being 1 square inch and the weight of a cubic inch of water .0361 pound. The volume of water discharged in 1 second is $1 \times 12 \times 20 = 240$ cubic inches, and since the weight of 1 cubic inch is .0361 pound, the weight discharged per second is $240 \times .0361 = 8.664$ pounds. The momentum of the stream is $\frac{8.664}{32.16} \times 20 = 5.38$ pounds. This is the constant force required to give a body of water weighing 8.664 pounds a velocity of 20 feet per second, in 1 second; it also represents the magnitude of the reaction on the nozzle, and the man must hold the nozzle with a force equal to the reaction, or 5.38 pounds, in order to prevent its slipping through his hands.

WORK, POWER, AND ENERGY.

WORK.

69. Work is the overcoming of resistance continually occurring along the path of motion.

Motion in itself is not work; a force must overcome a resistance in order that work may be done.

70. Unit of Work.—The unit by which the work done by a force is measured is the work done in overcoming a resistance of 1 pound through a space of 1 foot; this unit is called a **foot-pound.** According to the definition, it may be considered as the force required to raise 1 pound 1 foot vertically. All work is measured by this standard.

A horse going up a hill does an amount of work equal to its own weight, plus the weight of the wagon and its contents, plus the frictional resistances reduced to an equivalent weight, multiplied by the vertical height of the hill. Thus, if the horse weighs 1,200 pounds, the wagon and contents 1,200 pounds, and the frictional resistances equal 400 pounds, then if the vertical height of the hill is 100 feet, the work done is equal to (1,200 + 1,200 + 400) × 100 = 280,000 foot-pounds.

Rule 6.—*In all cases the force* (*or resistance*) *multiplied by the distance through which it acts equals the work. If a weight is raised, the weight multiplied by the vertical height of the lift equals the work.*

71. The total amount of work done in overcoming a given resistance through a given distance is independent of time; that is, it is immaterial whether it takes 1 minute or 1 year in which to do it; but in order to compare the *rate* at which work is done by different machines with a common standard, time must be considered. If one machine does a certain amount of work in 10 minutes and another machine does exactly the same amount of work in 5 minutes, the second machine can do twice as much work as the first in an equal period of time.

POWER.

72. **Power** is a term used to denote the rate at which work is done.

73. The common unit used for expressing the rate at which work is done is the **horsepower.**

One horsepower is 33,000 foot-pounds of work per minute; in other words, it is 33,000 pounds raised vertically 1 foot in 1 minute, or 1 pound raised vertically 33,000 feet in 1 minute, or any combination that will, when multiplied together, give 33,000 foot-pounds in 1 minute. Thus, the work done in raising 110 pounds vertically 5 feet in 1 second is a horsepower; for, since in 1 minute there are 60 seconds, 110 × 5 × 60 = 33,000 foot-pounds in 1 minute.

Example.—In a steam engine the force impelling the piston forwards and backwards is 10,000 pounds. This force overcomes the resistance due to the load at the rate of 600 feet per minute; that is, it moves the piston back and forth at that rate. What is the horsepower of the engine?

SOLUTION.—According to rule **6**, the work done is 10,000 × 600 = 6,000,000 foot-pounds per minute. Then, as 33,000 foot-pounds per minute represent a horsepower, the horsepower of the engine is 6,000,000 ÷ 33,000 = 181.818. Ans.

ENERGY.

74. Energy is a term used to express the ability of an agent to do work.

75. Kinetic Energy.—If we have a body at rest, a certain amount of force must be exerted and a certain amount of work must be done to set it in motion. A part of this force will be required to overcome those resistances outside of the body, such as friction and the resistance of the air, that oppose the motion of all bodies with which we have to do; another part acts to overcome the inertia of the body, to start it from its state of rest, and give it motion (see Newton's first law). The force that overcomes the resistance due to the inertia of the body does work, and the work so performed is stored in the body; in being brought to rest, the body is capable of overcoming a resistance and of doing an amount of work exactly equal to the work expended in giving it motion. The ability that the moving body has to do work while being brought to rest is called the **kinetic energy** of the body.

76. Rule for the Energy of a Moving Body.—

Let w = weight of body in pounds;
v = its velocity in feet per second;
E = kinetic energy in foot-pounds.

Then, the kinetic energy of a moving body may be found as follows:

Rule 7.—*Multiply the weight of the body by the square of its velocity and divide the product by twice the acceleration due to the force of gravity.*

Or, $$E = \frac{w\,v^2}{64.32}.$$

Thus, if a weight is raised a certain height, an amount of work is done equal to the product of the weight and the vertical height. If a weight is suspended at a certain height and allowed to fall, it will do the same amount of work in foot-pounds that was required to raise the weight to the height through which it fell.

EXAMPLE 1.—If a body weighing 25 pounds falls from a height of 100 feet, how much work can it do?

SOLUTION.—Work $= w\,h = 25 \times 100 = 2{,}500$ ft.-lb. Ans.

It requires the same amount of work or energy to stop a body in motion within a certain time as it does to give it that velocity in the same length of time.

EXAMPLE 2.—A body weighing 50 pounds has a velocity of 100 feet per second; what is its kinetic energy?

SOLUTION.—Kinetic energy $= \frac{w\,v^2}{64.32} = \frac{50 \times 100^2}{64.32} = 7{,}773.63$ ft.-lb. Ans.

EXAMPLE 3.—In the last example, how many horsepower will be required to give the body this amount of kinetic energy in 3 seconds?

SOLUTION.— 1 horsepower = 33,000 pounds raised 1 foot in 1 minute. If 7,773.63 foot-pounds of work are done in 3 seconds, in 1 second there will be done $\frac{7{,}773.65}{3} = 2{,}591.21$ foot-pounds of work. 1 horsepower = 33,000 foot-pounds per minute $= 33{,}000 \div 60 = 550$ foot-pounds per second.

The number of horsepower required will be

$$\frac{2{,}591.21}{550} = 4.7113 \text{ H. P.} \quad \text{Ans.}$$

77. Potential energy *is latent energy; it is the energy that a body at rest is capable of giving out under certain conditions.*

If a stone is suspended by a string from a high tower, it has potential energy. If the string is cut, the stone will fall to the ground, and during its fall its potential energy will change into kinetic energy, so that at the instant it strikes the ground its potential energy is wholly changed into kinetic energy.

At a point equal to one-half the height of the fall, the potential and kinetic energies are equal. At the end of the first quarter the potential energy is three-fourths and the kinetic energy one-fourth; at the end of the third quarter the potential energy is one-fourth and the kinetic energy three-fourths.

A pound of coal has a certain amount of potential energy. When the coal is burned, the potential energy is liberated and changed into kinetic energy in the form of heat. The kinetic energy of the heat changes water into steam, which thus has a certain amount of potential energy. The steam acting on the piston of an engine causes it to move through a certain space, thus overcoming a resistance, changing the potential energy of the steam into kinetic energy, and thus doing work.

Potential energy, *then, is the energy stored within a body that may be liberated and produce motion, thus generating kinetic energy and enabling work to be done.*

78. The principle of **conservation of energy** teaches that energy, like matter, can never be destroyed. If a clock is put in motion, the potential energy of the spring is changed into kinetic energy of motion, which turns the wheels, thus producing friction.

The friction produces heat, which dissipates into the surrounding air, but still the energy is not destroyed—it merely exists in another form.

79. Work of Acceleration and Retardation.—The theoretical amount of work that must be done in order to start a body from a state of rest and accelerate it until it reaches a given velocity is equal to the kinetic energy of the body at the given velocity. Likewise, the theoretical amount of work that must be done on a moving body to retard it and finally bring it to rest is equal to the kinetic energy the moving body possessed at the moment retardation began. The work that must be done in changing the velocity of a body is equal to the difference in the kinetic energies at the initial and final velocities. Since the motion of all bodies is opposed by some resisting force or forces, the actual amount of work required to give a body the given velocity will be the sum of the work of acceleration and the work required to overcome the outside resisting forces.

Example 1.—A body weighing 1,000 pounds is started from rest and is to attain a velocity of 88 feet per second in 2 minutes, passing over a distance of 5,280 feet in that time. If a constant force of 120 pounds must be exerted to overcome the frictional resistances, what work must be done?

Solution.—According to this article, the work required to accelerate the body is $\frac{1,000 \times 88^2}{64.32} = 120,398$ foot-pounds. As a constant force of 120 pounds must act through a distance of 5,280 feet to overcome the frictional resistances, the work done in overcoming friction is $5,280 \times 120 = 633,600$ foot-pounds. Then, the total amount of work done is $120,398 + 633,600 = 753,998$ ft.-lb. Ans.

Example 2.—What horsepower will be required to do the work calculated in the last example?

Solution.—As the work is done in 2 minutes, the horsepower is $\frac{753,998}{2 \times 33,000} = 11.424$ H. P., nearly. Ans.

FORCE OF A BLOW.

80. The questions are frequently asked, with what force will a falling hammer strike, or with what force will a projectile fired from a gun strike an object? These questions cannot be answered directly, as they are based on a misconception. A moving body possesses kinetic energy, or ability to do work, which ability can only be expressed in foot-pounds, but not in pounds of force, since the work done by the hammer or projectile in coming to rest is not a manifestation of force, but of energy.

Work is the product of force into distance; hence, if the amount of work a body has done or is capable of doing is known, the force can be determined for each case if, by some means, it is possible to determine exactly the distance in which the work is done. This distance depends on various resistances, such as that due to moving the object struck, the resistance to penetration, friction, the resistance to shearing or deformation of the body, etc. The distance through which these resisting forces act is generally indeterminate, and since the average of the resisting forces varies generally with the distance, this average resisting force is also indeterminate; hence, the force that, acting through a distance, will absorb all the kinetic energy of the hammer or projectile cannot be determined for the practical reasons given.

COMPOSITION AND RESOLUTION OF FORCES.

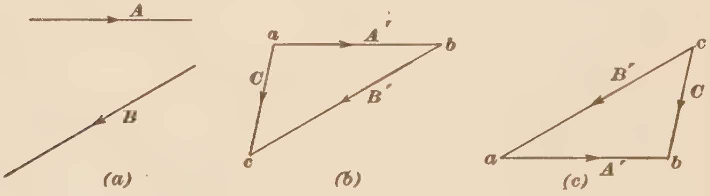
81. According to Art. **44**, in order that forces may be compared with one another, three conditions must be fulfilled. These conditions may all be represented by a line; hence, we may represent forces by lines. Thus, in Fig. 2, let *A* be the point of application of the force, let the length of the line *A B* represent its magnitude, and let the arrowhead indicate the direction in which the force acts; then, the line *A B* fulfils the three conditions and the force is fully represented.

FIG. 2.

COMPOSITION OF FORCES.

82. When two or more forces act upon a body at the same time along lines that meet in a common point, their combined effect on the body may be obtained by an application of the principle of the **triangle of forces.**

In Fig. 3 (*a*), let *A* and *B* be two forces having the magnitudes and directions represented by the two lines. To find the effect due to the combined action of these two forces, draw in any convenient location a line parallel to either of the



(*a*) (*b*) (*c*)

FIG. 3.

lines representing the two forces, making it equal in length, to some scale, to the magnitude of the force. Mark upon it an arrowhead pointing in the same direction as the arrowhead on the line representing the force. Then, from one extremity of the line just drawn draw a line parallel to the line representing the second force and equal in length, to the same scale, to the magnitude of the second force, and mark the arrowhead upon it. It is essential that the second line be so drawn that when passing over the two lines with a pencil, commencing at the beginning of either force, the arrowheads will both point in the same direction in reference to the direction of motion of the pencil.

The second line may be drawn from either end of the first line, but its direction must be made to fulfil the above absolutely essential condition. Thus, in Fig. 3 (*b*), the line B' has been drawn from the right-hand extremity of A'. Starting at a with a pencil and moving toward b, the pencil will move in the direction in which the arrowhead on A' points. Passing over B' from b to c, the pencil will move in the direction in which the arrowhead on B' points; we thus see that the lines are drawn correctly in reference to each other.

In Fig. 3 (*c*), the line B' has been drawn from the left-hand extremity of A'. Starting at c and following up the lines, it is seen that in this case the arrowheads both point in the same direction relative to the direction of motion of the pencil, thus showing the lines to be located correctly.

The two lines having been drawn, complete the triangle by drawing the line C. This line, called the **resultant,** represents the combined effect of the two forces; it gives the direction along which the two forces will act when combined. The magnitude of their combined effect is found by measuring this line by the scale with which A' and B' were laid off. The resultant will always have a direction opposite in sense to that of the forces; that is, if we pass a pencil around the triangle in the direction in which the arrowheads on the lines A' and B' point, the arrowhead on C, representing the direction of action of the resultant, must point in a direction opposite to that in which the pencil moves.

83. In practice it is often desired to find not only the magnitude and direction, but also the *actual location of the resultant*, the magnitudes and lines of action of the two forces being known. (By location of the forces and resultant is meant the location of the lines along which the forces actually act.) This can readily be done by producing the lines giving the location of the forces until they meet and then drawing the triangle of forces with their point of intersection as the starting point.

ILLUSTRATION.—In Fig. 4 is shown a head-frame erected at the mouth of a deep, vertical shaft. The hoisting rope that leads to the hoisting engine passes over a sheave at the top of the head-frame. A weight of 1,500 pounds hangs at the end of the rope. Neglecting the weight of the rope and sheave, what is the total pressure on the bearings of the sheave?

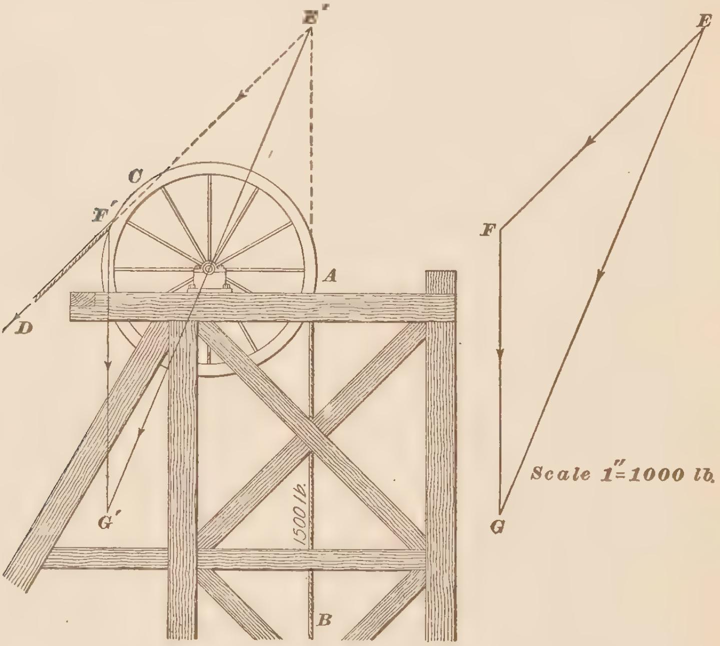


FIG. 4.

According to Art. **51**, the stress in both parts of the hoisting rope is 1,500 pounds. The part AB of the rope supports the weight;

consequently, the force acting along *A B* is downwards. The force acting along *C D* is the pull exerted by the engine and is toward the engine. To find the resultant of these two forces, draw the triangle of forces. Choosing a scale of 1 inch per 1,000 pounds, draw the line *E F* parallel to *C D*, making it $\frac{1500}{1000} = 1.5$ inches long. From *F* draw *F G* parallel to *A B* and $\frac{1500}{1000} = 1.5$ inches long. Join *E* and *G*; then *E G* will be the resultant whose magnitude is measured by the same scale. Measuring *E G*, it is found to be 2.75 inches long. As each inch represents 1,000 pounds, the magnitude of the resultant is $2.75 \times 1{,}000 = 2{,}750$ pounds.

To find the actual location of the resultant in reference to *A B* and *C D*, produce both lines, as shown in dotted lines, until they intersect at *E'*. Starting the triangle of forces at *E'*, lay off *E' F'* $= 1.5$ inches. From *F'* draw *F' G'* $= 1.5$ inches and parallel to *A B*. Join *E' G'*. Upon measurement it is found to be 2,750 pounds. As inspection shows that the resultant passes through the center of the bearings, the total pressure on the bearings is 2,750 pounds. Ans.

RESOLUTION OF FORCES.

84. Since two forces can be combined to form a single resultant force, we may also treat a single force as if it were the resultant of two forces whose action upon a body will be the same as that of a single force. Thus, in Fig. 5, the force *O A* may be resolved into two forces *O B* and *B A*. If the force *O A* acts upon a body moving or at rest upon a horizontal plane, and the resolved force *O B* is vertical and *B A* horizontal, *O B*, measured to the same scale as *O A*, is the magnitude of that part of *O A* that pushes the body *downwards*, while *B A* is the magnitude of that part of the force *O A* that is exerted in pushing the body in a *horizontal* direction. *O B* and *B A* are called the **components** of the force *O A*, and when these components are vertical and horizontal, as in the present case, they are called the *vertical component* and the *horizontal component* of the force *O A*.

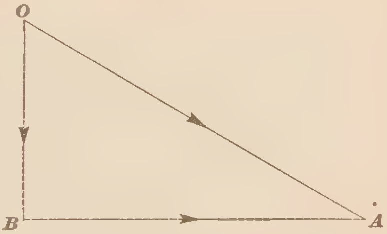


FIG. 5.

85. It frequently happens that the position, magnitude, and direction of a certain force is known, and that it is desired to know the effect of the force in some direction other than that in which it acts. Thus, in Fig. 5, suppose that *O A* represents, to some scale, the magnitude, direction, and line of action of a force acting upon a body at *A*, and that it is desired to know what effect *O A* produces in the direction *B A*. Now *B A*, instead of being horizontal, as in the figure, may have any direction. To find the value of the component of *O A* which acts in the direction *B A*, we use the following rule:

Rule 8.—*From one extremity of the line representing the given force, draw a line parallel to the direction in which it is desired that the component shall act; from the other extremity of the given force, draw a line perpendicular to the component first drawn, and intersecting it. The length of the component, measured from the point of intersection to the intersection of the component with the given force, will be the magnitude of the effect produced by the given force in the required direction.*

Thus, suppose *O A*, Fig. 5, represents a force acting upon a body resting upon a horizontal plane, and it is desired to know what *vertical pressure* *O A* produces on the body. Here the desired direction is vertical; hence, from one extremity, as *O*, draw *O B* parallel to the desired direction (vertical in this case), and from the other extremity draw *A B* perpendicular to *O B* and intersecting *O B* at *B*. Then *O B*, when measured to the same scale as *O A*, will be the value of the vertical pressure produced by *O A*.

86. Tangential Pressure.—One of the most familiar applications of the principle of the resolution of forces occurring in steam engineering is the case of the connecting-rod and crank. When the piston is at the end of its stroke and the crankpin is in a line drawn through the center of the cylinder and crank-shaft, a position that is expressed by saying the engine is "on the center," the pressure of the connecting-rod on the crankpin acts directly against the bearings of the shaft and there is no turning effect on the pin. After the pin leaves the center, the pressure exerted on it by the connecting-rod may be resolved into two components: One of these components acts in the direction of the center line PO of the crank (see Fig. 6) and merely exerts a pressure on the bearings of the shaft; the other component acts along the line AB at right angles to the center line PO and *tangent* to the circle described by the crankpin. This is the force that tends to turn the crank and is called the **tangential pressure** on the crankpin.

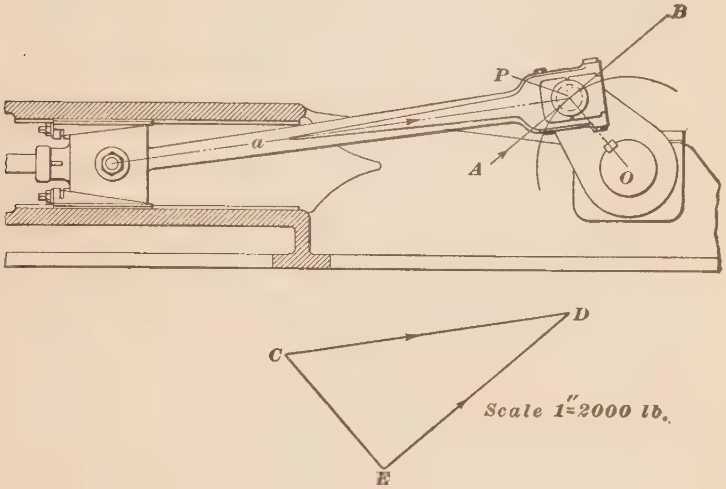


FIG. 6.

When the crank is on the center, there is no tangential pressure on the pin and no tendency to turn the crank. The tangential pressure gradually increases as the pin leaves the center and becomes greatest at the point where the connecting-rod and crank are at right angles to each other; it then decreases until the next center is reached. At the position where the connecting-rod and crank are at right angles to each other, the tangential pressure on the pin is equal to the total pressure exerted on it by the connecting-rod, and there is no component in the direction of the center line PO of the crank.

EXAMPLE.—If a force of 3,000 pounds acts along the connecting-rod *a* in the direction of the arrow (see Fig. 6), what is the tangential pressure on the crankpin?

SOLUTION.—As the tangential pressure is the pressure perpendicular to the crank, draw *A B* through the crankpin center at right angles to the center line of the crank; *A B* then represents the line along which the tangential pressure acts. Then, in any convenient location, draw *C D* parallel to the connecting-rod. Choosing a scale of 2,000 pounds = 1 inch, make the line *C D* 3,000 ÷ 2,000 = $1\frac{1}{2}$ inches long. From *D* draw an indefinite line *D E* parallel to *A B*, and draw *C E* perpendicular to *D E*. Now *E D*, measured to the scale adopted, will be the magnitude of the tangential pressure for the position of the crank and connecting-rod shown in the figure. Upon measurement, it is found to be 1.3 inches long. Then 1.3 × 2,000 = 2,600 lb., the tangential pressure. Ans.

87. When the total pressure on the piston rod of a steam engine is known, the force acting along the connecting-rod and the force acting upon the guides can be determined by the following application of the principle of the resolution of force:

Draw a line, as *A B*, Fig. 7, parallel to the line of motion of the crosshead *a*. Make its length, to some scale, equal in magnitude to the force impelling the piston. From one extremity of *A B* draw a line *B C* parallel to the center line of the connecting-rod *b*. From the other extremity of *A B* draw a line at a right angle to *A B*, producing it until it intersects the line *B C* at *D*. Then, *B D* represents the force acting along the connecting-rod and *A D* represents the force acting upon the guides.

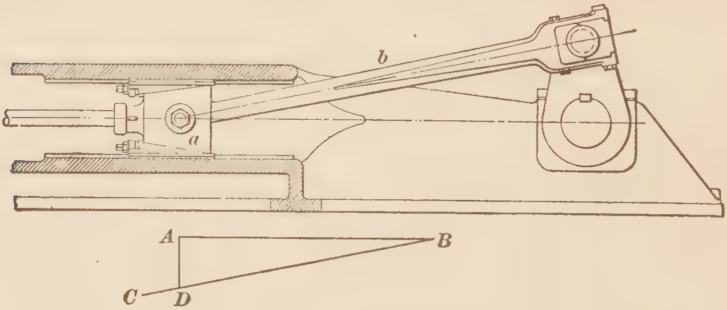


FIG. 7.

FRICTION.

88. Friction is the resistance that a body meets with from the surface upon which it moves.

89. The ratio between the *resistance* to the motion of a body due to friction and the *perpendicular* pressure between the surfaces is called the **coefficient of friction.**

If a weight W, as in Fig. 8, rests upon a horizontal plane, and has a cord fastened to it passing over a pulley a, from which a weight P is suspended, then, if P is just sufficient to start W, the ratio of P to W, or $\frac{P}{W}$, is the coefficient of friction between W and the surface it slides upon. The weight W is the perpendicular pressure, and P is the force necessary to overcome the resistance to the motion of W due to friction. If $W = 100$ pounds and $P = 10$ pounds, the coefficient of friction for this particular case would be $\frac{P}{W} = \frac{10}{100} = .1.$

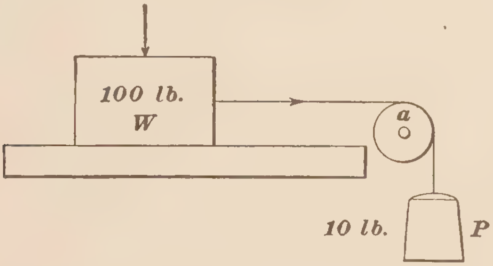


FIG. 8.

90. Laws of Friction.—

1. *Friction is directly proportional to the perpendicular pressure between the two surfaces in contact.*

2. *Friction is independent of the extent of the surfaces in contact when the total perpendicular pressure remains the same.*

3. *Friction increases with the roughness of the surfaces.*

4. *Friction is greater between surfaces of the same material than between those of different materials.*

5. *Friction is greatest at the beginning of motion.*

6. *Friction is greater between soft bodies than between hard ones.*

7. *Rolling friction is less than sliding friction.*

8. *Friction is diminished by polishing or lubricating the surfaces.*

91. Law 1 shows why the friction is so much greater on journals after they begin to heat than before. The heat causes the journal to expand, thus increasing the pressure between the journal and its bearing, and, consequently, increasing the friction.

Law 2 states that, no matter how small may be the surface that presses against another, if the perpendicular pressure is the same the friction will be the same. Therefore, large surfaces are used where possible, not to reduce the friction, but to reduce the wear and diminish the liability of heating.

For instance, if the perpendicular pressure between a journal and its bearing is 10,000 pounds, and the coefficient of friction is .2, the amount of friction is 10,000 × .2 = 2,000 pounds. Suppose that the area receiving the pressure is 80 square inches, then the amount of friction for each square inch is 2,000 ÷ 80 = 25 pounds.

If the area receiving the pressure had been 160 square inches, the friction would have been the same, that is, 2,000 pounds; but the friction per square inch would have been 2,000 ÷ 160 = 12½ pounds, just one-half as much as before, and the wear and liability to heat would be one-half as great also.

92. The values of the coefficient of friction given in the following tables are average values determined by General Morin, many years ago. Under certain conditions the coefficient may be considerably less than is given in the tables; it varies greatly, but the variation depends on so many conditions and the numerous experiments that have been made have given such contradictory results that no definite rules have yet been derived for determining the exact values under any condition. The student is, therefore, advised to use the values given in the tables, except where careful experiments have been made that give reliable values for the particular case under consideration. To find the force, in pounds, necessary to overcome friction, the coefficient taken from the table is multiplied by the perpendicular pressure, in pounds, on the surface considered. If the force acts at an angle to the surface, the perpendicular force can be found by resolving the given force into two components, one perpendicular and the other parallel to the surface.

TABLE I.

COEFFICIENTS OF FRICTION FOR PLANE SURFACES.

(*Reduced from M. Morin's Data.*)

Description of Surfaces in Contact.	State of the Surfaces.	Coefficient of Friction.
Wrought iron on cast iron.	Slightly greasy	.18
Wrought iron on bronze..	Slightly greasy	.18
Cast iron on cast iron....	Slightly greasy	.15
Cast iron on bronze......	Slightly greasy	.15
Bronze on bronze........	Dry	.20
Bronze on cast iron......	Dry	.22
Bronze on wrought iron..	Slightly greasy	.16
Cast iron, wrought iron, steel, and bronze sliding on one another or sliding on themselves.	Ordinary lubrication with lard, tallow, and oil	.07–.08
Cast iron, wrought iron, steel, and bronze sliding on one another or sliding on themselves.	Continuous lubrication	.05

TABLE II.

COEFFICIENTS OF FRICTION FOR JOURNAL FRICTION.

(*Reduced from M. Morin's Data.*)

Journals.	Bearings.	Lubricant.	Coefficient of Friction.	
			Ordinary Lubrication.	Continuous Lubrication.
Cast iron	Cast iron	Lard, olive oil, tallow	.07–.08	.03–.054
Cast iron	Bronze	Lard, olive oil, tallow	.07–.08	.03–.054
Wrought iron	Cast iron	Lard, olive oil, tallow	.07–.08	.03–.054
Wrought iron	Bronze	Lard, olive oil, tallow	.07–.08	.03–.054
Wrought iron	Lignum vitæ	Lard, olive oil	.11	
Bronze	Bronze	Oil	.10	
Bronze	Bronze	Lard	.09	

93. In the case of a weight sliding along a horizontal plane surface, the pressure is equal to the weight. When the surface is inclined, the weight acts vertically downwards, and the pressure perpendicular to the surface can be found by the principle of the resolution of forces. In many cases the pressure on the surfaces is due to the combined action of several forces that must be combined into one common resultant force.

94. The *work* that must be done in overcoming the resistance of friction depends on the distance through which the resistance is overcome. It may be calculated by the following rule:

Rule 9.—*Multiply the total pressure in pounds by the distance in feet and by the coefficient of friction.*

Let W = work in foot-pounds;
f = coefficient of friction;
p = total pressure in pounds;
d = distance in feet.

Then, $$W = p\,d\,f.$$

EXAMPLE.—The average perpendicular pressure on the guide of a steam engine due to the force impelling the piston is 2,500 pounds. The pressure due to the weight of the crosshead and connecting-rod is 400 pounds. The crosshead moves at the rate of 500 feet per minute; what horsepower is required to overcome the friction on the guides?

SOLUTION.—The total perpendicular pressure is $2{,}500 + 400 = 2{,}900$ pounds. Since the lubrication is usually intermittent, the coefficient of friction, for a brass slipper working on a cast-iron guide, may be taken as .08. The resistance being overcome through a distance of 500 feet each minute, the work done in overcoming friction is $2{,}900 \times 500 \times .08 = 116{,}000$ foot-pounds per minute. Then, the horsepower is $116{,}000 \div 33{,}000 = 3.52$ H. P., nearly. Ans.

95. In the case of a shaft rotating in a bearing, the distance through which friction is overcome each minute is found by multiplying the circumference of the journal by the number of revolutions per minute. For a shaft, or any other body rotating in a bearing, the force required to overcome friction, as calculated by multiplying the pressure by the coefficient of friction, is the force that must be applied *at the surface of the journal.*

96. Allowable Pressures.—It has been found by experience that when the pressure per unit of area exceeds a certain amount, the lubricant will be forced out and the bodies rubbing on each other will heat and, finally, seize.

The pressures that can safely be allowed on the bearings of steam engines, on guides, thrust bearings, crankpins, crosshead pins, etc. vary considerably, being dependent to a large extent on the character of the workmanship, the degree of finish, the variation of the pressure, the character of the lubrication, and the quality of the lubricant. With fair workmanship, the following pressures per square inch represent average practice in steam-engine work:

TABLE III.

PRESSURES PER SQUARE INCH ALLOWABLE IN STEAM-ENGINE WORK.

Engine Bearing.	Slow-Speed Stationary Engines. Pounds.	High-Speed Stationary and Marine Engines. Pounds.
Crankpins, iron......	800	400 to 600
Crankpins, steel......	1,200	
Wristpin.............	1,200	600 to 800
Main bearings.......	200	200
Guides	100	100
Thrust bearings......		60

97. For crankpins, wristpins, and guides, the allowable pressures given represent the pressures corresponding to the maximum load, which in the case of a wristpin and crankpin occurs when the crank, connecting-rod, and piston rod are in a straight line, and in the case of guides, when the connecting-rod and crank are at right angles to each other. In the case of pins and journals, the area to be considered in calculating the pressure on the bearing is the **projected area,** which is found by multiplying the length of the journal by its diameter.

EXAMPLES FOR PRACTICE.

1. A body weighs 90 pounds; what is its mass? Ans. 2.799, nearly.

2. What force will be required to accelerate a body at the rate of 2 feet per second, the body weighing 450 pounds, and the frictional resistances being equal to 10 per cent. of the weight of the body?
Ans. 72.99 lb., nearly.

3. What work is done in raising 950 pounds 17 feet?
Ans. 16,150 ft.-lb.

4. If an engine does 205,000 foot-pounds of work per minute, what is its horsepower? Ans. 6.21 H. P., nearly.

5. What is the kinetic energy of a shell fired from a cannon with a velocity of 1,800 feet per second, the shell weighing 1,000 pounds?
Ans. 50,373,135 ft.-lb., nearly.

6. Taking the coefficient of friction at .15, what horsepower will be required to pull 100 pounds at a uniform speed of 5 feet per second along a level surface?
Ans. .136 H. P.

CENTER OF GRAVITY.

98. *The center of gravity of a body is that point at which the body may be balanced, or it is the point at which the whole weight of a body may be considered as concentrated.*

This point is not always *in* the body; in the case of a horseshoe or a ring it lies outside of the substance of, but within the space enclosed by, the body.

In a moving body, the line described by its center of gravity is always taken as the path of the body. In finding the distance that a body has moved, the distance that the center of gravity has moved is taken.

The definition of the center of gravity of a body may be applied to a system of bodies if they are considered as being connected at their centers of gravity.

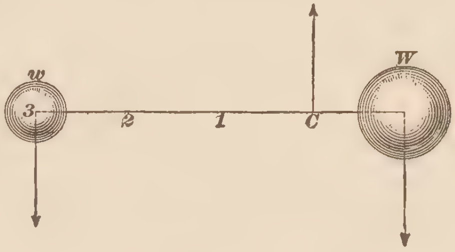


FIG. 9.

If w and W, Fig. 9, are two bodies of known weight, their center of gravity will be at C. This point C may be readily determined as follows:

Rule 10.—*The distance of the common center of gravity from the center of gravity of the large weight is equal to the weight of the smaller body multiplied by the distance between the centers of gravity of the two bodies, and this product divided by the sum of the weights of the two bodies.*

EXAMPLE.—In Fig. 9, $w = 10$ pounds, $W = 30$ pounds, and the distance between their centers of gravity is 36 inches; where is the center of gravity of both bodies situated?

Solution.—Applying the rule, $10 \times 36 = 360$. $10 + 30 = 40$. $360 \div 40 = 9$ in., distance of center of gravity from center of large weight. Ans.

99. It is now very easy to extend this principle to find the center of gravity of any number of bodies, when their weights and the distances apart of their centers of gravity are known, by the following rule:

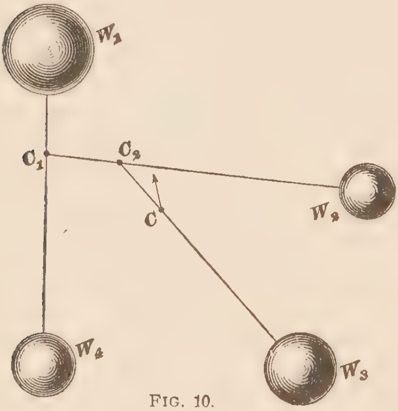


Fig. 10.

Rule 11.—*Find the center of gravity of two of the bodies, as* W_1 *and* W_4 *in Fig. 10. Assume that the weight of both bodies is concentrated at* C_1, *and find the center of gravity of this combined weight at* C_1 *and the weight of* W_2. *Let it be at* C_2; *then find the center of gravity of the combined weights of* W_1, W_4, W_2 *(concentrated at* C_2*), and* W_3. *Let it be at* C; *then* C *will be the center of gravity of the four bodies.*

100. To find the center of gravity of **any parallelogram:**

Rule 12.—*Draw the two diagonals, Fig. 11, and their point of intersection C will be the center of gravity.*

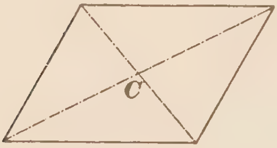


Fig. 11.

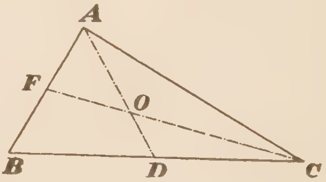


Fig. 12.

101. To find the center of gravity of a **triangle,** as *A B C*, Fig. 12:

Rule 13.—*From any vertex, as A, draw a line to the middle point D of the opposite side $B\,C$. From one of the other vertexes, as C, draw a line to F, the middle point of the opposite side $A\,B$; the point of intersection O of these two lines is the center of gravity.*

It is also true that the distance $D\,O = \frac{1}{3}\,D\,A$ and that $F\,O = \frac{1}{3}\,F\,C$; the center of gravity could have been found by drawing from any vertex a line to the middle point of the opposite side and measuring back from that side $\frac{1}{3}$ of the length of the line.

The center of gravity of **any regular plane figure** is the same as the center of the inscribed or circumscribed circle.

102. To find the center of gravity of **any irregular plane figure,** but of uniform thickness throughout, divide one of the parallel surfaces into triangles, parallelograms, circles, ellipses, etc., according to the shape of the figure; find the area and center of gravity of each part separately, and combine the centers of gravity thus found in the same

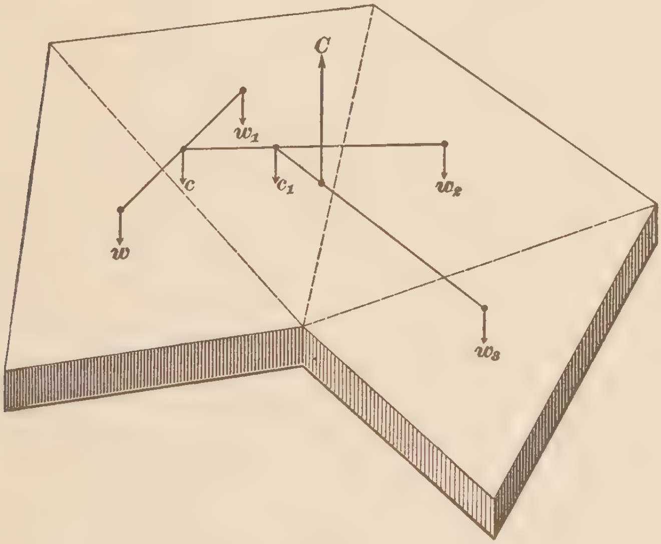


FIG. 13.

manner as in rule **11**, in this case, however, dealing with the *area* of each part instead of its weight. See **Fig. 13.**

103. Center of Gravity of a Solid.—In a body free to move, the center of gravity will lie in a vertical plumb-line drawn through the point of support. Therefore, to find the position of the center of gravity of an irregular solid, as the crank, Fig. 14, suspend it at some point, as *B*, so that it will move freely. Drop a plumb-line from the point of suspension, and mark its direction. Suspend the body at another point, as *A*, and repeat the process. The intersection *C* of the two lines will be directly over the center of gravity.

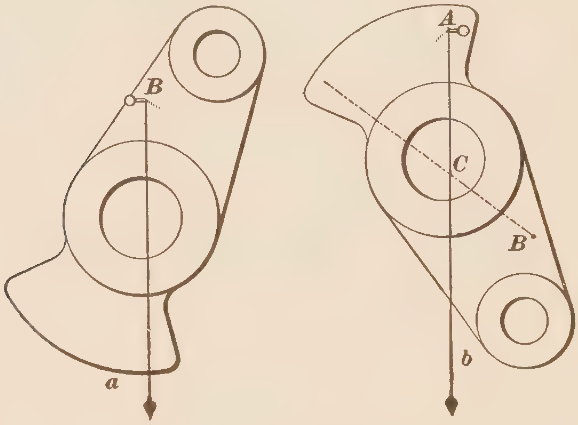


FIG. 14.

It is often desired to find the horizontal distance of the center of gravity from a given point of the body. In many cases this can readily be done by balancing the body on a knife edge, and then measuring, horizontally, the distance between the knife edge and the given point.

Since the position of the center of gravity depends wholly on the shape and weight of a body, it may be without the body, as in the case of a circular ring, whose center of gravity is the same as the center of the circumference of the ring.

By considering the symmetry of form, the position of the center of gravity of homogeneous solids may often be determined without analysis, or it may be limited to a certain plane, line, or point. Thus, the center of gravity of a sphere, or any other regular body, is situated at its center; of a cylinder, in the middle of its axis; of a thin plate having the form of a circle or regular polygon, in the center of the figure; of a straight wire of uniform cross-section, in the middle of its length.

EXAMPLES FOR PRACTICE.

1. A spherical shell has a wrought-iron handle attached to it. The shell is 10 inches in diameter and weighs 20 pounds. The handle is 1½ inches in diameter, and the distance from the center of the shell to the end of the handle is 4 feet. Where is the center of gravity? Take the weight of a cubic inch of wrought iron as .278 pound.

Ans. 13.612 in. from center of shell.

2. The distance between the centers of two bodies is 51 inches. The weights of the bodies being 20 and 73 pounds, where is the center of gravity? Ans. 10.968 in. from the center of the larger weight.

3. Weights of 5, 9, and 12 pounds lie in one straight line in the order named. Distance from the 5-pound weight to the 9-pound weight is 22 inches, and from the 9-pound weight to the 12-pound weight is 18 inches. Where is the center of gravity?

Ans. 13.923 in. from 12-lb. weight.

CENTRIFUGAL FORCE.

104. If a body is fastened to a string and whirled, so as to give it a circular motion, there will be a pull on the string that will be greater or less, according as the velocity increases or decreases. The cause of this pull on the string will now be explained.

Suppose that the body is revolved horizontally, so that the action of gravity upon it will always be the same. According to the first law of motion, a body put in motion tends to move in a straight line unless acted upon by some other force, causing a change in the direction. When the body moves in a circle, the force that causes it to move

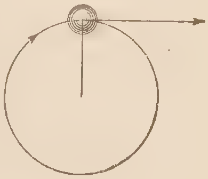


FIG. 15.

in a circle instead of in a straight line is exactly equal to the tension of the string. If the string were cut, the pulling force that drew it away from the straight line would be removed, and the body would then "fly off at a tangent"; that is, it would move in a straight line tangent to the circle, as shown in Fig. 15.

Since, according to the third law of motion, every action has an equal and opposite reaction, we call the force that acts as an equal and opposite force to the pull of the string the **centrifugal force,** and it acts *away* from the center of motion.

105. The other force, or tension, of the string is called the **centripetal force,** and it acts *toward* the center of motion. It is evident that these two forces, acting in opposite directions, tend to pull the string apart, and, if the velocity be increased sufficiently, the string will break. It is also evident that no body can revolve without generating centrifugal force.

106. To Find the Centrifugal Force of Any Revolving Body.—The value of the centrifugal force, expressed in pounds, of any revolving body is calculated by the following rule:

Rule 14.—*The centrifugal force equals the continued product of .00034, the weight of the body in pounds, the radius in feet (taken as the distance between the center of gravity of the body and the center about which it revolves), and the square of the number of revolutions per minute.*

Let $F =$ centrifugal force in pounds;
$W =$ weight of revolving body in pounds;
$R =$ radius in feet of circle described by center of gravity of revolving body;
$N =$ revolutions per minute of revolving body.

Then, $$F = .00034\, W R N^2.$$

Example.—What is the tension in the string of Fig. 15, if the ball weighs 2 pounds and is revolved around at the rate of 500 revolutions per minute? The string is 2 feet long.

Solution.—Applying the rule just given, we get

$$F = .00034 \times 2 \times 2 \times 500^2 = 340 \text{ lb. Ans.}$$

107. In flywheels, belt wheels, and pulleys the centrifugal force tends to tear the rim asunder; this tendency is resisted by the tenacity of the material of which the wheel is composed. Since the centrifugal force increases as the square of the number of revolutions, it will be seen that an apparently slight increase in the number of revolutions per minute may be sufficient to burst the wheel.

108. For solid cast-iron wheels and for built-up wheels of cast iron where the strength of the joint is equal to the strength of the rim, the greatest number of revolutions per minute that practice has indicated to be safe may be found by the following rule:

Rule 15.—*Divide 1,930 by the diameter of the wheel.*

Or,

$$N = \frac{1{,}930}{d};$$

where d = diameter of the wheel in feet;
N = number of revolutions per minute.

Example.—What is the maximum number of revolutions allowable for a cast-iron flywheel 27 feet 6 inches in diameter?

Solution.—Applying rule **15**, we get

$$N = \frac{1{,}930}{27.5} = 70 \text{ rev. per min. Ans.}$$

EQUILIBRIUM.

109. When a body is at rest, all the forces that act upon it balance one another and are said to be in **equilibrium.** The most important force to be considered is the attraction of the earth, which acts upon every particle of a body. There are three states of equilibrium: *stable*, *unstable*, and *neutral.*

110. A body is in **stable equilibrium** when, if slightly rotated about its support in such manner as to change the elevation of its center of gravity, it tends to return to its position of rest. Examples of bodies in a state of stable equilibrium are a cube resting on one of its sides, a cone resting on its base, a pendulum, etc. A body can only be in stable equilibrium when a rotation about its support raises the center of gravity.

111. A body is in **unstable equilibrium** when a rotation about its support, so as to change the elevation of its center of gravity, tends to make it fall farther from its position of rest. Examples of bodies in unstable equilibrium are a cube balanced on one of its edges, a cone standing upon its point, an egg balanced upon its end, etc. When a body is in unstable equilibrium, any rotation, no matter how slight, tends to lower its center of gravity.

112. A body is in **neutral equilibrium** when a rotation about its support does not change the elevation of its center of gravity. Examples of bodies in neutral equilibrium are a sphere of uniform density and a cone resting on its side.

113. A vertical line drawn through the center of gravity of a body is called the line of direction. So long as the line of direction falls within the base, the body will stand; when the line of direction falls outside of the base, the body will fall.

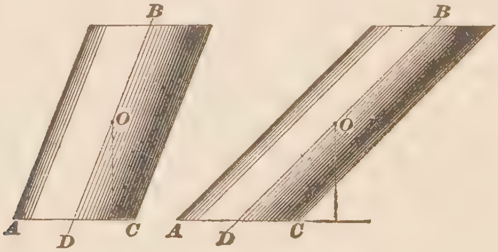


FIG. 16.

Let $A\,C\,B$, Fig. 16, be a cylinder whose base is oblique to the center line $B\,O\,D$, and let O be the center of gravity of the cylinder. So long as the vertical line drawn through *O* falls between *A* and *C*, the cylinder will stand, but the instant it falls without the base, the cylinder will fall.

114. The center of gravity of a body has a tendency to always seek the lowest possible position.

MACHINE ELEMENTS.

THE LEVER, WHEEL, AND AXLE.

FUNDAMENTAL PRINCIPLES.

1. A **lever** is a bar capable of being turned about a pivot, or point, as in Figs. 1, 2, and 3.

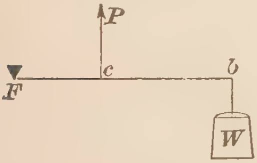


FIG. 1.

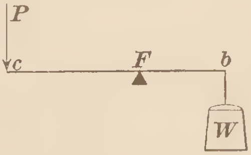


FIG. 2.

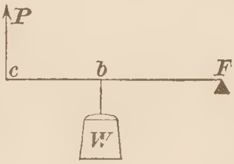


FIG. 3.

The object W to be lifted is called the **weight**; the force is represented by P*; and the point, or pivot F, is called the **fulcrum.**

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* The force applied to a lever, screw, wheel and axle, or any similar machine element in order to raise a given weight, was formerly called the *power*, and the arm of a lever to which the force is applied was called the *power arm*; at present, however, the term power is universally used to represent the rate at which work is done, and, hence, its application to those cases where a simple force is meant often leads to a serious confusion of ideas regarding the relation between force, work, and power. To prevent this confusion, we will use the word power only in accordance with the definition given.

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2. That part of the lever $F\,b$ between the fulcrum and the weight is called the **weight arm,** and the part $F\,c$ between the fulcrum and the force is called the **force arm.**

In order that the lever will be in equilibrium (balance), *the force multiplied by the force arm must equal the weight multiplied by the weight arm;* that is, $P \times F\,c$ must equal $W \times F\,b$.

3. If F is taken as the center of a circle, and arcs are described through b and c, it will be seen that if the weight arm is moved through a certain angle, the force arm will move through the same angle. Since in the same or equal angles the lengths of the arcs are proportional to the radii with which they were described, it is seen that the force arm is proportional to the distance through which the force acts, and the weight arm is proportional to the distance through which the weight moves. Hence, instead of writing $P \times F\,c = W \times F\,b$, we might have written $P \times$ (distance through which P acts) $= W \times$ (distance through which W moves). This is the general law of all machines, and can be applied to any mechanism from the simple lever up to the most complicated arrangement. When stated in the form of a rule it is as follows:

Rule 1.—*The force multiplied by the distance through which it acts equals the weight multiplied by the distance through which it moves.*

4. In the above rule, it will be noticed that there are four requirements necessary for a complete knowledge of the lever, viz.: the force, the weight, the force arm, or distance through which the force acts, and the weight arm, or distance through which the weight moves. If any three are given, the fourth may be found by letting x represent the requirement that is to be found, and multiplying the force by the force arm and the weight by the weight arm; then, dividing the product of the two known numbers by the number by which x is multiplied, the result will be the requirement that is to be found.

EXAMPLE.—If the weight arm of a lever is 6 inches long and the force arm is 4 feet long, how great a weight can be raised by a force of 20 pounds at the end of the force arm?

SOLUTION.—In this example, the weight is unknown; hence, representing it by x, we have, after reducing the 4 feet to inches, $20 \times 48 = 960$ = force multiplied by the force arm, and $x \times 6$ = weight multiplied by the weight arm. Dividing the 960 by 6, the result is 160 pounds, the weight. Ans.

5. If the distance through which the force acted or the weight had moved had been given instead of the force arm or weight arm, and it were required to find the force or weight, the process would have been exactly the same, using the given distance instead of the force arm or weight arm.

EXAMPLE.—If, in the above example, the weight had moved $2\frac{1}{2}$ inches, through what distance would the force have acted?

SOLUTION.—In this example, the distance through which the force acts is required. Let x represent the distance. Then, $20 \times x$ = distance multiplied by force, and $2\frac{1}{2} \times 160 = 400$ = distance multiplied by the weight. Hence, $x = \frac{400}{20} = 20$ inches = distance through which the force arm moves. Ans.

The ratio between the weight and the force is $160 \div 20 = 8$. The ratio between the distance through which the weight moves and the distance through which the force acts is $2\frac{1}{2} \div 20 = \frac{1}{8}$. This shows that while a force of 1 pound can raise a weight of 8 pounds, the 1-pound weight must move through 8 times the distance that the 8-pound weight does. It will also be noticed that the ratio of the lengths of the two arms of the lever is also 8, since $48 \div 6 = 8$.

6. The law that governs the straight lever also governs the bent lever, but care must be taken to determine the true lengths of the lever arms, which are, in every case, *the perpendicular distances from the fulcrum to the line of direction of the weight or force.*

Thus, in Figs. 4, 5, 6, and 7, $F c$ in each case represents the force arm and $F b$ the weight arm.

7. A **compound lever** is a series of single levers arranged in such a manner that when a force is applied to the first it is communicated to the second, and from that to the third, and so on.

Fig. 8 shows a compound lever. It will be seen that when a force is applied to the first lever at *P* it will be communicated to the second lever at *P*, from this to the third lever at *P*, and thus raise the weight *W*.

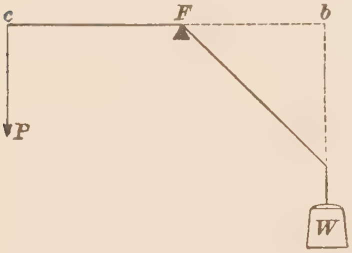

FIG. 4.

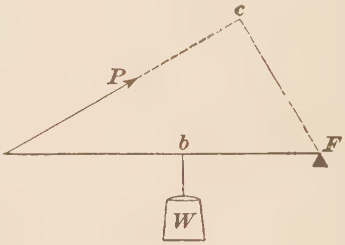

FIG. 5.

The weight that the force applied to the first lever could raise acts as the force of the second, and the weight that this force could raise through the second lever acts as the force of the third lever, and so on, no matter how many single levers make up the compound lever.

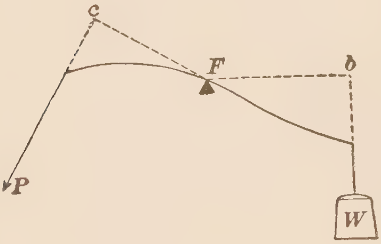

FIG. 6.

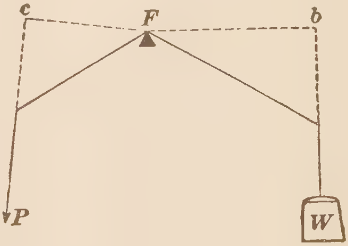

FIG. 7.

In this case, as in every other, the force multiplied by the distance through which it acts equals the weight multiplied by the distance through which it moves.

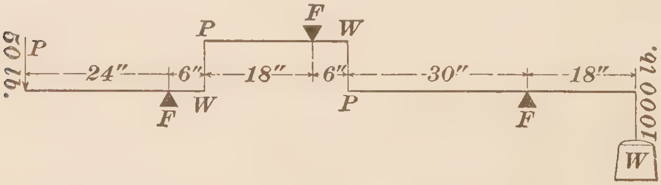

FIG. 8.

Hence, if we move the P end of the lever, say 4 inches, and the end carrying the weight W moves $\frac{1}{5}$ inch, we know that the ratio between P and W is the same as the ratio between $\frac{1}{5}$ and 4; that is, 1 to 20, and, hence, that 10 pounds at P will balance 200 pounds at W, without measuring the lengths of the different lever arms. If the lengths of the lever arms are known, the ratio between P and W may be readily obtained from the following rule:

Rule 2.—*The continued product of the force and each force arm equals the continued product of the weight and each weight arm.*

EXAMPLE.—If, in Fig. 8, PF = 24 inches, 18 inches, and 30 inches, respectively, and WF = 6 inches, 6 inches, and 18 inches, respectively, how great a force at P will it require to raise 1,000 pounds at W? What is the ratio between W and P?

SOLUTION.—Let x represent the force; then, $x \times 24 \times 18 \times 30 = 12{,}960 \times x$ = continued product of the force and each force arm. $1{,}000 \times 6 \times 6 \times 18 = 648{,}000$ = continued product of the weight and each weight arm; and, since $12{,}960 \times x = 648{,}000$,

$$x = \frac{648{,}000}{12{,}960} = 50 \text{ lb.} = \text{the force. Ans.}$$

$$1{,}000 \div 50 = 20 = \text{ratio between } W \text{ and } P. \text{ Ans.}$$

8. The **wheel and axle** consists of *two cylinders of different diameters rigidly connected*, so that they turn

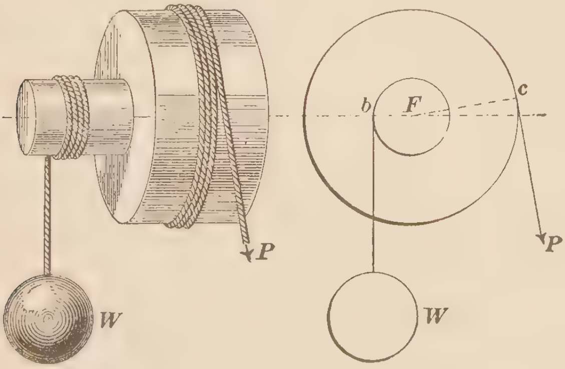

FIG. 9.

together about a common axis, as in Fig. 9. Then, as before, $P \times$ distance through which it acts $= W \times$ distance through which it moves; and, since these distances are proportional to the radii of the force cylinder and weight cylinder, $P \times Fc = W \times Fb$.

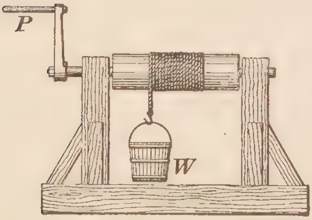

FIG. 10.

It is not necessary that an entire wheel be used; an arm, projection, radius, or anything that the force causes to revolve in a circle, may be considered as the wheel. Consequently, if it is desired to hoist a weight with a windlass, Fig. 10, the force is applied to the handle of the crank, and the distance between the center line of the crank-handle and the axis of the drum corresponds to the radius of the wheel.

EXAMPLE.—If the distance between the center line of the handle and the axis of the drum in Fig. 10 is 18 inches and the diameter of the drum is 6 inches, what force will be required at P to raise a load of 300 pounds?

SOLUTION.— $P \times (18 \times 2) = 300 \times 6$, or $P = 50$. Ans.

EXAMPLES FOR PRACTICE.

1. The lever of a safety valve is of the form shown in Fig. 1, where the force is applied at a point between the fulcrum and the weight lifted. If the distance from the fulcrum to the valve is $5\frac{1}{2}$ inches and from the fulcrum to the weight is 42 inches, what total force is necessary to raise the valve, the weight being 78 pounds and the weight of valve and lever being neglected? Ans. 595.64 lb.

2. If, in Fig. 8, $PF = 10$ inches, 12 inches, 14 inches, and 16 inches, respectively, and $WF = 2$ inches, 3 inches, 4 inches, and 5 inches, respectively, (*a*) how great a weight can a force of 20 pounds raise? (*b*) What is the ratio between W and P? (*c*) If P moves 4 inches, how far will W move?

$$\text{Ans.} \begin{cases} (a) & 4{,}480 \text{ lb.} \\ (b) & 224. \\ (c) & \frac{1}{56} \text{ in.} \end{cases}$$

3. A windlass is used to hoist a weight. If the diameter of the drum on which the rope winds is 4 inches, and the distance from the center of the handle to the axis of the drum is 14 inches, how great a weight can a force of 32 pounds applied to the handle raise?

Ans. 224 lb.

PULLEYS.

9. Pulleys for the transmission of power by belts may be divided into two principal classes: (1) The *solid pulley* shown in Fig. 11, in which the hub, arms, and rim are one entire casting. (2) The *split pulley* shown in Fig. 12, which is cast in halves.

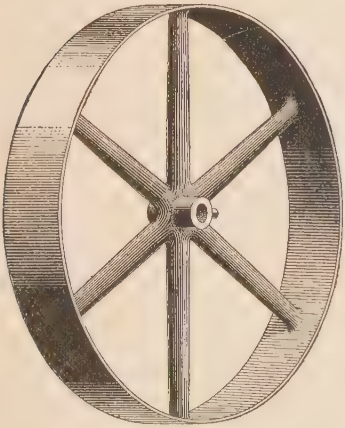

FIG. 11.

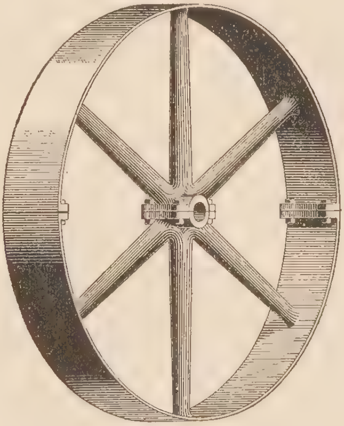

FIG. 12.

The latter style of pulley is more readily placed on and removed from the shaft than the solid pulley. Pulleys are generally cast in halves or parts when they are more than 6 feet in diameter; this is done on account of the shrinkage strain in large pulley castings, which renders them liable to crack as a result of the unequal cooling of the metal.

10. Of late years, wooden pulleys have come into extensive use. They are built of segments securely glued together, maple being the wood used. Wooden split pulleys can be procured that are fitted with removable bushings, thus allowing the same pulley to be readily adapted to various diameters of shafting. They are somewhat lighter than cast-iron pulleys.

11. Crowning Pulley Faces.—In Fig. 13, suppose the shafts *a* and *b* to be parallel. Let the pulley on the shaft *a* be cone-shaped, as shown. The right-hand side of the belt will be pulled ahead more rapidly than the left-hand side, because of the greater diameter and, consequently, greater speed of the right-hand end of the pulley. It has been observed that in this case the belt will leave its normal position, which is indicated by the dotted lines, and climb toward the part of the pulley that has the largest diameter, as shown in the illustration. This tendency of the belt to climb toward the high side is taken advantage of to make the belt stay on a pulley. Suppose the pulley on the shaft *a* is replaced with one formed of two equal cones, with the large diameter in the center. Then, each side of the belt will tend to climb toward the highest point, and the consequence is that the belt will stay in the center of the pulley.

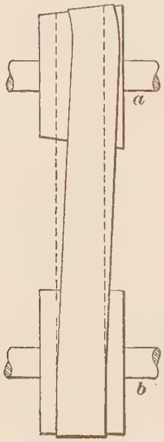

FIG. 13.

A pulley with its surface formed in this way is said to be **crowned.** The surface need not necessarily represent the frustums of two cones; it may simply be curved, as shown in Fig. 14. It is only required that the pulley be larger in diameter in the center.

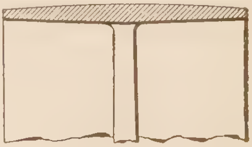
FIG. 14.

As to the proper amount of crowning necessary, the practice of the makers of pulleys differs considerably; usually, though, it is from $\frac{3}{16}$ to $\frac{1}{2}$ inch per foot of width of the pulley face.

12. Balancing Pulleys.—All pulleys that rotate at high speeds should be balanced. If they are not, the centrifugal force that is generated by the rotation of the pulley, on account of the center of gravity being outside the axis of rotation, will cause vibration; hence, pulleys are usually balanced in the manner shown in Fig. 15. A closely fitting arbor, which is simply a truly cylindrical piece of iron or steel, is driven into the bore of the pulley, which is then placed on the so-called "balancing ways." These are two planed iron or steel bars, preferably planed to a knife edge. These bars are placed on convenient supports far enough apart to allow the pulley to go between them. The bars should be carefully leveled with a spirit level, so that both bars are in the same horizontal plane. When the arbor rests on the ways and the pulley is slightly rotated, the latter will quickly come to rest with the heavy part downwards. Now, either some metal must be removed from the heavy part or some weight added to the light part. For small pulleys, a convenient substance to use for finding the proper location and weight of the counterweight is ordinary glazier's putty. By repeated trials the proper weight of the counterweight is found, and then a mass of metal of convenient shape equal in weight to the putty is fastened to the light part of the pulley in whatever way is safe and convenient. The proper weight and location of the counterweight will have been obtained when the pulley will be at rest on the balancing ways in any position in which it is put. In other words, the pulley is balanced when it is in neutral equilibrium. This balance is called a **standing balance.** The method just explained answers very well for pulleys that are narrow in comparison

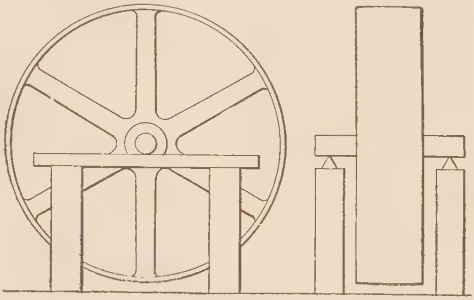
FIG. 15.

to their diameter. When pulleys with a wide face are run at a high speed, it is often found that serious vibrations are set up even when they are in perfect standing balance, thus showing that they do not possess the so-called **running balance.** No method has yet been found that will insure a running balance at all speeds, nor has any method become known by which it can be discovered directly where to apply the counterweight. The usual remedy for pulleys not possessing a running balance is to turn carefully the inside of the rim to run true with the outside of the pulley. Absence of running balance is, in all cases, due to an unequal distribution of metal in reference to the axis of the shaft, this unequal distribution being due to lack of homogeneity of the metal, to poor foundry work, or to poor lathe work.

13. Pulleys should run true in order that the strain, or tension, of the belt will be equal at all parts of the revolution, thus making the transmitting power equal. The smoother the surface of a pulley, the greater is its driving power.

The transmitting power of a pulley can be increased by covering its face with a leather or rubber band; this increases the driving power about one-quarter.

14. Relation Between Speeds of Drivers and Driven Pulleys.—The pulley that imparts motion to the belt is called the **driver**; that which receives the motion is called the **driven.**

The revolutions of any two pulleys over which a belt is run vary in an inverse proportion to their diameters; consequently, if a pulley 20 inches in diameter is driven by one 10 inches in diameter, the 20-inch pulley will make 1 revolution while the 10-inch pulley makes 2 revolutions, or they are in the ratio of 2 to 1. From this fact, the following formulas have been deduced:

Let $D =$ diameter of the driver;
$d =$ diameter of the driven;
$N =$ number of revolutions of the driver;
$n =$ number of revolutions of the driven.

NOTE.—The words revolutions per minute are frequently abbreviated to R. P. M.

15. To find the diameter of the driving pulley when the diameter of the driven pulley and the number of revolutions per minute of each is given:

Rule 3.—*The diameter of the driving pulley equals the product of the diameter and the number of revolutions of the driven pulley divided by the number of revolutions of the driving pulley.*

Or, $$D = \frac{d\,n}{N}.$$

EXAMPLE.—The driving pulley makes 100 revolutions per minute, the driven pulley makes 75 revolutions per minute and is 18 inches in diameter; what is the diameter of the driving pulley?

SOLUTION.—Applying the rule just given and substituting, we have $D = \frac{18 \times 75}{100} = 13\frac{1}{2}$ in. Ans.

16. The diameter and number of revolutions per minute of the driving pulley being given, to find the diameter of the driven pulley, which must make a given number of revolutions per minute:

Rule 4.—*The diameter of the driven pulley equals the product of the diameter and the number of revolutions of the driving pulley divided by the number of revolutions of the driven pulley.*

Or, $$d = \frac{D\,N}{n}.$$

EXAMPLE.—The diameter of the driving pulley is 13½ inches and it makes 100 revolutions per minute; what must be the diameter of the driven pulley to make 75 revolutions per minute?

SOLUTION.—Applying rule 4 we have $d = \frac{13\frac{1}{2} \times 100}{75} = 18$ in. Ans.

17. To find the number of revolutions per minute of the driven pulley, its diameter and the diameter and the number of revolutions per minute of the driving pulley being given:

Rule 5.—*The number of revolutions of the driven pulley equals the product of the diameter and the number of revolutions of the driver divided by the diameter of the driven pulley.*

Or,
$$n = \frac{D N}{d}.$$

Example.—The driving pulley is $13\frac{1}{2}$ inches in diameter and makes 100 revolutions per minute; how many revolutions will the driven pulley make in 1 minute if it is 18 inches in diameter?

Solution.—Formula, $n = \frac{D N}{d}$.

Substituting, we have $n = \frac{13\frac{1}{2} \times 100}{18} = 75$ R. P. M. Ans.

18. To find the number of revolutions per minute of the driving pulley, its diameter and the diameter and the number of revolutions per minute of the driven pulley being given:

Rule 6.—*The number of revolutions of the driving pulley equals the product of the diameter and the number of revolutions of the driven pulley divided by the diameter of the driving pulley.*

Or,
$$N = \frac{d n}{D}.$$

Example.—The driven pulley is 18 inches in diameter and makes 75 revolutions per minute; how many revolutions will the driving pulley make in 1 minute if it is $13\frac{1}{2}$ inches in diameter?

Solution.—Formula, $N = \frac{d n}{D}$.

Substituting, we have $N = \frac{18 \times 75}{13\frac{1}{2}} = 100$ R. P. M. Ans.

BELTS.

19. A **belt** is a flexible connecting band that drives a pulley by its frictional resistance to slipping at the surface of the pulley. Belts are most commonly made of leather, cotton, or rubber, and are united in long lengths by cementing, riveting, or lacing.

20. Leather belts are made single and double. A single belt is one composed of a single thickness of leather; a double belt is one composed of two thicknesses of leather cemented and riveted together the whole length of the belt.

21. Cotton belts are in use to some extent, as are also belts made of a number of layers of duck sewed together and impregnated with a preparation rendering them waterproof. These belts, in accordance with the number of layers or "plys," are called two-ply, three-ply, etc. Four-ply cotton and duck belting is about equal to single leather belting, and eight-ply to double leather belting.

22. Rubber belts are especially adapted for use in damp or wet places; they will endure a great degree of heat or cold without injury, are quite durable, and are claimed to be less liable to slip than leather belts.

CALCULATIONS FOR BELTS.

23. To Find the Length of a Belt.—In practice, the necessary length for a belt to pass around pulleys that are already in their position on a shaft is usually obtained by passing a tape line around the pulleys, the stretch of the tape line being allowed as that necessary for the belt. The lengths of open-running belts for pulleys not in position can be obtained approximately as follows:

Rule 7.—*The length of a belt for open-running pulleys equals $3\frac{1}{4}$ times one-half the sum of the diameters of the pulleys plus 2 times the distance between the centers of the shaft.*

Let D = diameter of one pulley in inches;
d = diameter of the other pulley in inches;
L = distance between the centers of the shafts in inches;
B = length of the belt in inches.

Then,
$$B = 3\tfrac{1}{4}\left(\frac{D+d}{2}\right) + 2L.$$

Example.—The distance between the centers of two shafts is 9 feet 7 inches; the diameter of the large pulley is 36 inches and the diameter of the small one is 14 inches; what is the necessary length of the belt?

Solution.—Applying the rule just given and substituting the values given, we have, since 9 feet 7 inches = 115 inches,

$$B = 3\tfrac{1}{4}\left(\frac{36 + 14}{2}\right) + 2 \times 115 = 311\tfrac{1}{4} \text{ in., or } 25 \text{ ft. } 11\tfrac{1}{4} \text{ in. Ans.}$$

The length of crossed belts cannot be determined by any simple calculation, it being a rather difficult mathematical problem.

24. To Find the Width of Belts.—A belt should be wide enough to bear safely and for a reasonable length of time the greatest tension that will be put upon it. This will be the tension of the driving side. The safe tension for single belts may be taken as 60 pounds per inch of width; single belts average $\frac{3}{16}$ inch in thickness. The tension on the driving side, however, does not represent the force tending to turn the pulley. The force tending to turn the pulley, or the **effective pull,** is the difference in tension between the driving side and the slack side of the belt. The tension on the driving side depends on three factors: the *effective pull* of the belt, the *coefficient of friction* between the belt and pulley, and the size of the *arc of contact* of the belt on the smaller pulley.

25. The effective pull that may be allowed per inch of width for single leather belts with different arcs of contact (the arc in which the belt touches the smaller pulley), is given in Table I.

26. To Find the Arc of Contact.—The arc of contact in degrees, or as a fraction of the circumference, can be determined, practically, as follows: Stretch a string over the two pulleys to represent the belt. Then, take another string, wrap it around the small pulley and cut it off so that the ends meet. This represents the circumference of the small pulley. Now take a third string, hold one end at the

TABLE I.

ALLOWABLE BELT PULLS.

Arc Covered by Belt.		Allowable Effective Pull Per Inch of Width in Pounds.
Degrees.	Fraction of Circumference.	
90	.250	23.0
112½	.312	27.4
120	.333	28.8
135	.375	31.3
150	.417	33.8
157½	.437	34.9
180 or over	.500	38.1

beginning of the arc of contact, as shown by the string stretched around both pulleys, wrap it around the smaller pulley, and cut it off at the end of the arc of contact. The length of this last string represents the length of the arc of contact. We now have the proportion: *the length of the string representing the circumference* : *the length of the string representing the arc of contact* :: *360* (*the number of degrees in a circle*) : *the number of degrees in the arc of contact.* Whence, the number of degrees in the arc of contact equals the quotient obtained by dividing the product of the length of the arc of contact and 360 by the circumference of the pulley.

To obtain the fraction of the circumference, divide the length of the string representing the arc of contact by the circumference.

27. To use the table for finding the width of a single leather belt for transmitting a given number of horsepower, we have the following rule,

where C = allowable effective pull, from table;
H = horsepower to be transmitted;
W = width of single belt in inches;
V = velocity of belt in feet per minute.

Rule 8.—*Multiply the horsepower to be transmitted by 33,000, and divide this product by the product of the velocity of the belt and the allowable effective pull, as taken from the table. The quotient will be the width of the belt.*

Or,
$$W = \frac{33{,}000\,H}{V\,C}.$$

Example.—What width of single belt is needed to transmit 20 horsepower, the arc of contact on the small pulley being 135° and the speed of the belt 1,500 feet per minute?

Solution.—According to Table I, the allowable effective pull for 135° is 31.3 pounds. Then, applying rule **8,** we have

$$W = \frac{33{,}000 \times 20}{1{,}500 \times 31.3} = 14 \text{ in., nearly.} \quad \text{Ans.}$$

28. To Find the Horsepower of a Belt.—The horsepower that a single belt will transmit is given by the following rule:

Rule 9.—*Multiply together the effective pull taken from the table, the width of the belt in inches, and the speed of the belt in feet per minute. Divide the product by 33,000.*

Or,
$$H = \frac{C\,W\,V}{33{,}000}.$$

29. Speed of Belts.—By applying rule **9** to the same belt running at different velocities, it will be seen that the higher the velocity, the greater is the horsepower that the same belt can transmit, and from rule **8** it will be seen that the higher the speed of the belt, the less may be its width to transmit a given horsepower. From this it follows that a belt should be run at as high a velocity as conditions will permit, the maximum velocity allowable for a laced belt being about 3,500 feet per minute for ordinary single leather and double leather belts. For belts spliced by cementing, where the splice is practically as strong as the belt itself, the velocity may be as high as 5,000 feet per minute. Cases are on record where wide main belts have been run at as high a velocity as 6,000 feet per minute.

30. In choosing a proper belt speed, due regard must be paid to commercial conditions. While a high speed of the belt means a narrow and, consequently, a cheaper belt, the increased cost of the larger pulleys that may be required may offset the gain due to the high belt speed, at least as far as first cost is concerned.

For illustration, let the problem be to transmit 10 horsepower from one shaft to another. Let the revolutions of both the driven and the driving shaft be equal, and let the shafts make 200 revolutions per minute. Choosing a belt speed of 2,000 feet per minute, the width of a single belt to transmit the given horsepower will be, by rule **8**, say, $4\frac{1}{2}$ inches. The diameter of the pulley that at 200 revolutions per minute will give a belt speed of 2,000 feet per minute is $\frac{2,000 \times 12}{200 \times 3.1416} = 38\frac{1}{4}$ inches, nearly. Taking the price of a 38-inch cast-iron pulley, 5-inch face, as $15, we have the price of two pulleys as $30. Let the distance between the pulleys be 20 feet. Then, the length of belt, according to rule **7**, is 50 feet 4 inches. Taking 51 feet as the length of the belt, and the price of a single leather belt $4\frac{1}{2}$ inches wide at 50 cents per foot, the price of the belt will be $25.50. Then, the cost of belt and pulleys, not counting freight or express charges, etc., will be $25.50 + $30 = $55.50.

Choosing a belt speed of 3,500 feet per minute, the width of belt will be $2\frac{1}{2}$ inches, nearly. The proper diameter of the pulley is $\frac{3,500 \times 12}{200 \times 3.1416} = 66\frac{7}{8}$ inches, say, 67 inches. The length of the belt will be 59 feet, about. The price of the belt at 30 cents per foot is $17.70. The price of two 67-inch pulleys $3\frac{1}{2}$-inch face is, say, $80. Then, the total first cost is $80 + $17.70 = $97.70, showing that in this particular case the use of a low belt speed reduces the first cost by \$97.70 – \$55.50 = \$42.20.

The above illustration is not intended, and must not be construed, to be an argument against high belt speed; it simply shows the advisability of considering the commercial features in each and every case. In many cases it will be found that the narrow, high-speed belt is by far the more economical one to use.

31. Double belts are made of two single belts cemented and, usually, riveted together their whole length, and are used where much power is to be transmitted. As the effective pull for single belts, as given in Table I, is based primarily on the strength through the lace holes, a double belt, which is twice as thick, should be able to transmit twice as much power as a single belt, and in fact more than this, where, as is quite common, the ends of the belt are cemented instead of laced.

Where double belts are used on small pulleys, however, the contact with the pulley face is less perfect than it would be if a single belt were used, owing to the greater rigidity of the former. More work is, also, required to bend the belt as it runs over the pulley than in the case of the more pliable single belt, and the centrifugal force tending to throw the belt from the pulley also increases with the thickness. Moreover, in practice, it is seldom that a double belt is put on with twice the tension of a single belt. For these reasons, the width of a double belt required to transmit a given horsepower is generally assumed to be seven-tenths the width of a single belt to transmit the same power. On this basis, rules **8** and **9** become for double belts, by multiplying rule **8** by $\frac{7}{10}$ and dividing rule **9** by $\frac{7}{10}$, as follows:

Rule 10.—*To find the width of a double belt, multiply the horsepower to be transmitted by 23,100. Divide this product by the product of the velocity of the belt and the allowable effective pull, as taken from the table.*

Or, $$W = \frac{23{,}100\,H}{V\,C}.$$

Rule 11.—*To find the horsepower that a double belt can transmit, multiply together the effective pull taken from the table, the width of the belt, and its velocity. Divide the product by 23,100.*

Or, $$H = \frac{C\,W\,V}{23{,}100}.$$

Example 1.—What width of double belt is required to transmit 20 horsepower, the arc of contact on the smaller pulley being 135° and the speed of the belt 1,500 feet per minute?

Solution.—According to Table I, the effective pull is 31.3 pounds. Then, by rule **10**,

$$W = \frac{23{,}100 \times 20}{1{,}500 \times 31.3} = 10 \text{ in., nearly.} \quad \text{Ans.}$$

Example 2.—What horsepower can be transmitted by a 6-inch double belt running at 400 feet per minute with an arc of contact of 180°?

Solution.—According to Table I, the effective pull is 38.1 pounds. Applying rule **11**, we have

$$H = \frac{38.1 \times 6 \times 400}{23{,}100} = 4 \text{ H. P., nearly.} \quad \text{Ans.}$$

THE CARE AND USE OF BELTS.

32. Leather Belts.—It is a much disputed question as to which side of the belt should be run next to the pulley. The more common practice, it is believed, is to run the belt with the hair or grain side nearest the pulley. This side is harder and more liable to crack than the flesh side. By running it on the inside the tendency is to cramp or compress it as it passes over the pulley, while if it ran on the outside, the tendency would be for it to stretch and crack. The flesh side is the tougher side, but for the reason given above the life of the belt will be longer if the wear comes upon the grain side. The lower side of the belt should be the driving side, the slack side running from the top of the driving pulley. The sag of the belt will then cause it to encompass a greater length of the circumference. Long belts, running in any other direction than vertical, work better than short ones, as their weight holds them more firmly to their work.

It is bad practice to use rosin to prevent slipping. It gums the belt, causes it to crack, and prevents slipping for only a short time. If a belt, in good condition, persists in slipping, a wider belt should be used. Sometimes larger pulleys on the driving and driven shafts are of advantage, as they increase the belt speed and reduce the stress on the belt. Belts may be kept soft and pliable by oiling them once a month with castor oil or neatsfoot oil.

33. The Flapping of Belts.—One of the most annoying troubles experienced with belting of all kinds is the violent flapping of the slack side. Flapping may be due to any one of several causes, or to a combination of them. The most usual cause is that one or both of the pulleys run out of true. The belt is then alternately stretched and released, and while this may not cause flapping at one speed, it will usually do so at a higher speed. If the belt is rather slack, tightening it somewhat may cure or alleviate the flapping. The most obvious and best remedy, but the most expensive, is to turn the pulleys to run true.

Pulleys being out of line with each other are another prolific source of flapping, especially when one or both run out of true. First, bring the pulleys in line; if this fails, tighten the belt if it is rather loose. If no improvement is noticed and it is not possible to turn the pulleys, try to lower the belt speed a little, either by the substitution of smaller pulleys or by changing the speed of the driving shaft, according to circumstances.

With belts running at speeds above 4,000 feet per minute, flapping may occur when the pulleys are perfectly true and in line with each other, even when the belt has the proper tension. This is believed to be due to air becoming entrapped between the face of the pulley and the belt. At any rate, it has been observed that perforating the belt with a series of small holes will cure this trouble. Perforated belts may now be bought in the market.

Lack of steadiness in running, due either to sudden variations in the speed of the engine, or sudden changes in the load of the machines driven by the belt, will produce a flapping that it is almost impossible to cure. The only known cure is to take such steps as will insure steady running, as for instance, increasing the weight of the flywheel on the engine, or placing a flywheel instead of a pulley on the driven machine.

The belt not being joined square will also cause flapping, especially when the belt is running at a rather high speed. The remedy is to unlace or unfasten the joint and make it square.

Too great a distance between the pulleys may also cause flapping. In general, the distance between the pulleys should not exceed 15 feet for belts up to 4 inches in width; 20 feet for belts above 4 and below 12 inches; 25 feet for belts above 12 inches and below 18 inches; and 30 feet for larger belts. The distances here given are occasionally exceeded considerably, but as no experiments have ever been made public that would enable a fairly correct formula to be deduced for the distance between pulleys when the belt speed, width of belt, and effective pull are known, they must be taken as representing average practice.

A horizontal belt is, by many, considered to have the proper tension when it has about 1 inch of sag, while in motion, for every 8 feet between the pulleys. For belts other than horizontal, this should be less, there being no sag at all for vertical belts.

JOINING THE ENDS OF BELTS.

34. Lacing.—The ends of a belt may be joined by *lacing*, *sewing*, *riveting*, or *cementing*. Many ways of **lacing** belts are used. A very satisfactory method for belts up to 3 inches in width is shown in Fig. 16. Cut the ends of the belt square, using a sharp knife and a try square. Punch a row of holes according to the width of the belt, punching corresponding holes exactly opposite each other in each end of the belt, using 3 holes in belts up to 2 inches wide, and 5 holes in belts between 2 and 3 inches wide. The number of holes in the row should always be uneven for the style of lacing shown. In the figure, *A* is the outside of the belt and *B* the side running nearest the pulley. The lacing should be drawn halfway through one of the middle holes from the under side, as at *1*; before going any further, it is well to see to it that the belt has no twists in it, or, in the case of a crossed belt, that it has not been given a wrong twist. The same side of the belt should run over both pulleys, which will be the case with a crossed belt if it has been twisted correctly. Having made sure that the belt is fair, pass the end of the lace on the upper side of the belt through *2* under the belt and up through *3*, back again through *2* and *3*, through *4* and up through *5*, where an incision is made in one side of the lacing, which forms a barb that will prevent the end from pulling through. Lace the right-hand side in the same manner. The lacing may advantageously be carried on at once to the right and left alternately.

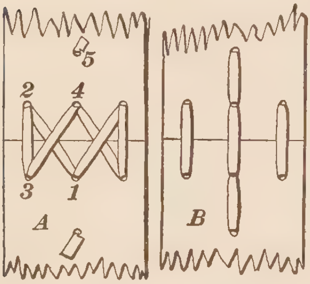

FIG. 16.

35. For belts wider than 3 inches, the lacing shown in Fig. 17 is a good one. As will be observed, there are two rows of holes. The number of holes in the row nearest the joint should exceed by one the number of holes in the second row. For belts up to 4½ inches wide, use 3 holes in the row nearest the joint and 2 holes in the second row. For belts up to 6 inches wide, use 4 and 3 holes, respectively. For larger belts, make the total number of holes in each end either one or two more than the number of inches of width, with the object of getting an odd total number of holes. For example, for a 10-inch belt, the total number of holes should be $10 + 1 = 11$. For a 13-inch belt, it should be $13 + 2 = 15$ holes. The outside holes of the first row should not be nearer the edges of the belt than ¾ inch, nor should the first row be nearer the joint than ⅞ inch. The second row should be at least 1¾ inches from the end. In the figure, *A* is the outside and *B* the side nearest the pulley. Begin at one of the center holes in the outside row, as *1*, and continue through *2*, *3*, *4*, *5*, *6*, *7*, *6*, *7*, *4*, *5*, *2*, *3*, etc.

Another method is to begin the lacing on one side instead of in the middle. This method will give the rows of lacing on the under side of the belt the same thickness all the way across. The lacing should not be crossed on the side of the belt that runs next to the pulley.

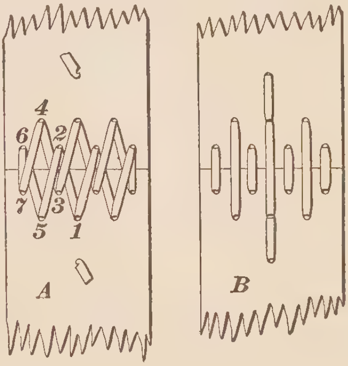

FIG. 17.

Lacing affords convenient means of *shortening belts* when they stretch and of *increasing their tension*. If the belts are at all large, the ends of the belt should be drawn together by belt clamps.

36. Cementing.—Cementing makes probably the best kind of a joint ever devised. It has the serious disadvantage, however, that the stretch of the belt cannot be readily taken up, and, hence, tightening pulleys must be used when the center-to-center distance of the pulleys is not adjustable.

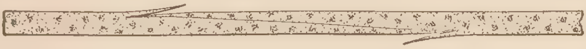

FIG. 18.

In dynamo driving, where endless belts are used to the exclusion of all others, the dynamo is usually mounted on a slide, so that the tension of the belt can be adjusted. For a cemented joint, the ends of the belt should be pared down with a very sharp knife to the form shown in Fig. 18, which shows a form that is recommended by belt manufacturers.

Warm the belt ends near a fire, apply the belt cement while hot, and press the joint together by two boards, one on the top and one on the bottom. Belt cement can be obtained of any dealer in engineer's supplies and full directions for using it will always be found on the can. These directions should be implicitly followed.

If belt cement cannot be obtained, a good cement may be made by melting together over a slow fire 16 parts of gutta percha, 4 parts of india rubber, 2 parts of pitch, 1 part of shellac, and 2 parts of linseed oil, by weight. Cut all ingredients very small, mix well, and use while hot.

37. Stretch of Belts.—New leather belts will stretch from one-fourth to one-half of an inch per foot of length, and hence must be taken up until the limit of stretch has been reached. Rubber belts are said to stretch continuously. Cotton and duck belts are said not to stretch with use.

38. Precautions to be Observed When Using Rubber Belts.—When rubber belts are used, animal oils or animal grease should never be used on them. If the belt should slip, it should be lightly moistened on the side nearest the pulley with boiled linseed oil.

EXAMPLES FOR PRACTICE.

1. The main line shaft is driven at 90 revolutions per minute and is to drive another shaft at 120 revolutions per minute; the latter shaft carries a pulley 20 inches in diameter. How large a pulley should be used on the main line shaft? Ans. 26⅔ in.

2. The belt wheel of an engine is 10 feet in diameter and makes 65 revolutions per minute; the line shaft is to run at 150 revolutions per minute. What size pulley should be used? Ans. 52 in.

3. The driving pulley is 48 inches in diameter and makes 90 revolutions per minute. The driven pulley being 20 inches in diameter, how many revolutions per minute will the driven shaft make? Ans. 216 R. P. M.

4. The belt wheel of an engine being 5 feet in diameter, the driven pulley 45 inches, and the driven shaft to make 90 revolutions per minute, what should be the number of revolutions of the belt wheel? Ans. 67½ R. P. M.

5. Two pulleys, 48 and 32 inches in diameter, respectively, are 10 feet 6 inches from center to center. What is the length of an open belt for these pulleys? Ans. 31 ft. 10 in.

6. A belt speed of 2,400 feet per minute having been chosen, what width of single belt will be needed to transmit 30 horsepower? The pulleys over which the belt runs are equal in size. Ans. 11 in., nearly.

7. What horsepower can be transmitted by a single belt 11 inches wide traveling at the rate of 1,000 feet per minute over a small pulley on which its arc of contact is 150°? Ans. $11\frac{1}{4}$ H. P., nearly.

8. With a belt speed of 1,300 feet per minute, what width of double belt will be required to transmit 5 horsepower over pulleys equal in diameter? Ans. $2\frac{1}{2}$ in., nearly.

WHEEL WORK.

DRIVER AND FOLLOWER.

39. A combination of wheels and axles, as in Fig. 19, is called a train. The wheel in a train to which motion is imparted from a wheel on another shaft, by such means as a belt or gearing, is called the **driven wheel** or **follower**; the wheel that imparts the motion is called the **driver.**

It will be seen that the wheel and axle bears the same relation to the train that a simple lever does to the compound lever. Letting D_1, D_2, D_3, etc. represent the diameters of the driven wheels, and d_1, d_2, d_3, etc. the diameters of the different drivers, we have the following rule:

Rule 12.—*The continued product of the force and the diameters of the driven wheels equals the continued product of the weight, the diameter of the drum that moves the weight, and the diameters of the drivers.*

Or, $P \times D_1 \times D_2 \times D_3$, etc. $= W \times d_1 \times d_2 \times d_3$, etc.

Since the radii of several circles are in the same proportion to one another as the diameters of the same circles, it follows that the radii may be used instead of the diameters in the above rule.

EXAMPLE.—If the radius of the pulley A, Fig. 19, is 20 inches, of C 15 inches, and of E 24 inches, and the radius of the drum F is 4 inches, of the pinion D 5 inches, and of the pinion B 4 inches, how great a weight will a force of 1 pound applied at P raise?

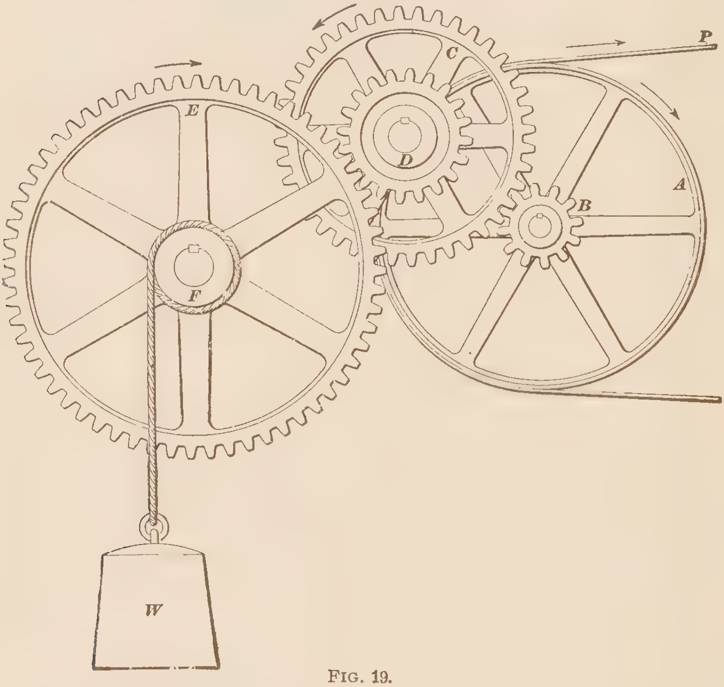

FIG. 19.

SOLUTION.—Applying rule **12**, we have

$$1 \times 20 \times 15 \times 24 = W \times 4 \times 5 \times 4,$$

or

$$W = \frac{7,200}{80} = 90 \text{ lb.} \quad \text{Ans.}$$

40. Although the combination of wheels in this example enables the lifting of a weight 90 times as great as the force applied, it has been necessary to exert the force through a distance 90 times the height through which the weight was raised. It is a universal law in the application of machines that *whenever there is a gain in power without a corresponding increase in the initial force, there is a loss in speed;* this is true of any machine.

In the example, if P were to move the entire 90 inches in 1 second, W would move only 1 inch in the same period of time.

41. Instead of using the diameter or radius of a gear, as in the last example, the number of teeth may be used when computing the weight that can be raised, or the velocity.

EXAMPLE.—The radius of the pulley A, Fig. 19, is 40 inches and that of F is 12 inches. The number of teeth in B is 9, in C 27, in D 12, and in E 36. If the weight to be lifted is 1,800 pounds, how great a force at P is it necessary to apply to the belt?

SOLUTION.—Let P represent the force; then, by the rule

$$P \times 40 \times 27 \times 36 = 1{,}800 \times 12 \times 9 \times 12,$$

or

$$P \times 38{,}880 = 2{,}332{,}800.$$

Hence,

$$P = \frac{2{,}332{,}800}{38{,}880} = 60 \text{ lb.},$$

= the amount of force necessary to apply to the belt. Ans.

GEAR-WHEELS.

42. A wheel that is provided with teeth that mesh with similar teeth on another wheel is called a **gear-wheel,** or **gear.** In Fig. 20 is shown a **spur gear.** On spur gears

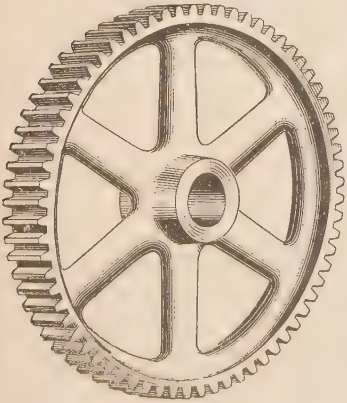

FIG. 20.

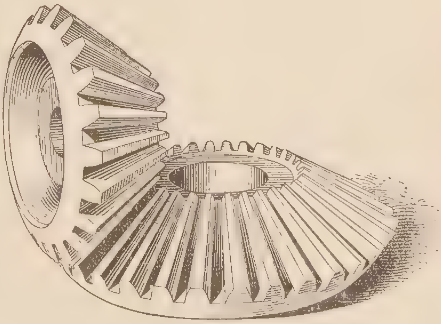

FIG. 21.

the teeth are usually parallel to the axis of the wheel or to its shaft.

Gears are said to be *in mesh* when the teeth of two wheels, respectively, engage each other or interlock.

43. In Fig. 21 is shown a pair of **bevel gears** in mesh. Of the two wheels shown, one is smaller than the other; when both wheels of a pair of bevel gears are of the same diameter they are called **miter gears.** In Fig. 22 is shown a pair of miter gears in mesh. It is obvious that the angle that the teeth of these gears make with the axis of the shaft must be 45°.

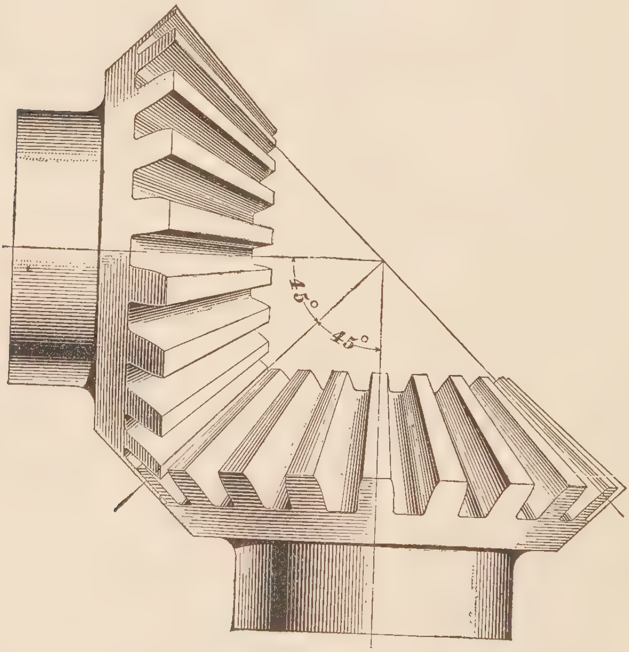

FIG. 22.

44. Of a pair of gear-wheels (either spur or bevel) having different diameters, the smaller is called a **pinion.**

In Fig. 23 is shown a revolving screw, or **worm,** as it is called, that meshes with a **worm-wheel.** It is used to transmit motion from one shaft to another at right angles to it.

As the worm is nothing else than a screw, each revolution given to it will rotate the wheel a distance equal to the pitch of the worm; consequently, if there are 40 teeth in the worm-wheel, a single-threaded worm must make 40 revolutions in order to turn the wheel once.

45. A **rack** is a straight bar that has gear-teeth cut on it. It may be considered as part of a gear-wheel in which the diameter is infinitely large. The teeth of racks are proportioned by the same rules as those of gear-wheels.

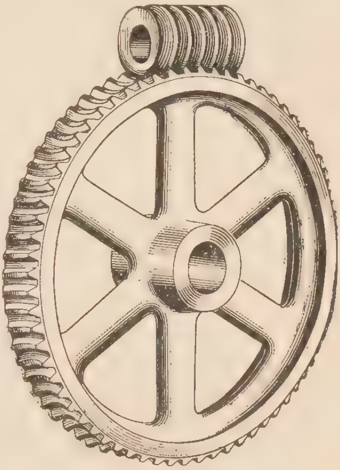
FIG. 23.

TEETH OF GEAR-WHEELS.

46. The object in designing the teeth of gear-wheels should be to so shape them that the motion transmitted will be exactly the same as with a corresponding pair of wheels, or cylinders, without teeth and running in contact without slipping. Such cylinders are called **pitch cylinders,** and are always represented on the drawing of a gear-wheel by a line called the **pitch circle** (see Fig. 24). The pitch circle is also called the **pitch line.**

The diameter of the pitch circle is called the **pitch diameter.** When the word "diameter" is applied to gears, it is always understood to mean the *pitch diameter*, unless especially stated as the "diameter over all" or "diameter at the root."

47. Circular and Diametral Pitch.—The distance from a point on one tooth to a corresponding point on the next tooth, *measured along the pitch circle*, is the **circular pitch.** This is obtained by dividing the circumference (pitch circle) by the number of teeth, and is used in laying out the teeth of large gears and, also, when calculating their strength.

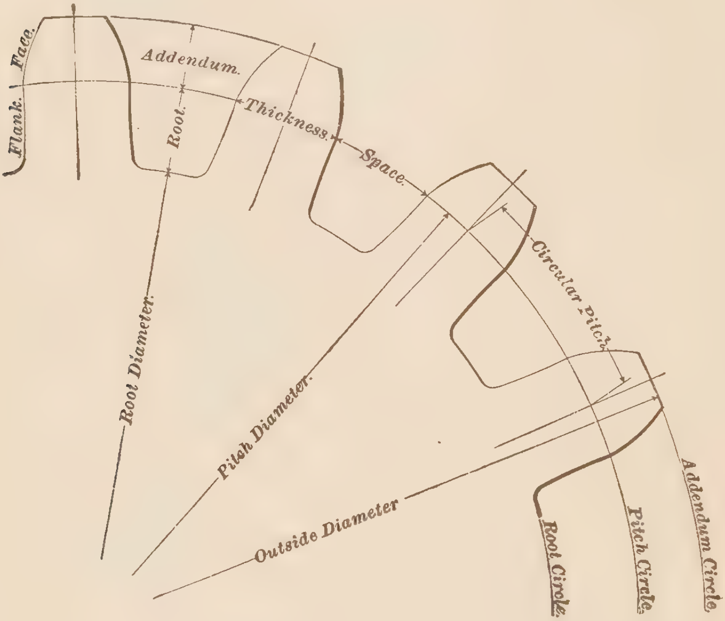

FIG. 24.

It would be very convenient to have the circular pitch expressed in manageable numbers like 1-inch, $\frac{3}{4}$ inch, etc.; but as the circumference of a gear is 3.1416 times its diameter, this requires awkward numbers for the diameters. Thus, a wheel of 40 teeth, 1-inch pitch, would have a circumference on the pitch circle of 40 inches and a diameter of 12.732 inches. Of the two it is more convenient, in the great majority of cases, to have the diameters expressed in numbers that can be easily handled. In order, however, to have the pitch in a convenient form also, a pitch has been devised that is *expressed in terms of the diameter* and called the **diametral pitch.**

The diametral pitch is not a measurement like the circular pitch, but a *ratio. It is the ratio of the number of teeth in the gear to the number of inches in the diameter; or, it is the number of teeth on the circumference of the gear for 1 inch diameter of the pitch circle.* It is obtained by dividing the number of teeth by the diameter.

A gear, for example, has 60 teeth and is 10 inches in diameter. The diametral pitch is the ratio of 60 to $10 = \frac{60}{10} = 6$; this gear would be called a 6-pitch gear. From the definition it follows that teeth of any particular diametral pitch are of the same size and have the same width on the pitch line, whatever may be the diameter of the gear. Thus, if a 12-inch gear has 48 teeth, it will be 4 pitch. A 24-inch gear to have teeth of the same size will have twice 48, or 96 teeth, and as $96 \div 24 = 4$, has the same diametral pitch as before.

48. Other Definitions.—The other necessary definitions applying to the parts of a gear can be readily understood from Fig. 24. The thickness of the tooth and the width of the space are measured on the pitch circle. A tooth is composed of two parts, the **addendum,** or part outside of the pitch circle, and the **root,** which is inside.

A line through the outside end of the addendum is called the **addendum circle,** or **addendum line,** and one through the inside part of the root is called the **root circle,** or **root line.** The amount by which the width of the space is greater than the thickness of the tooth is called the **backlash,** or **side clearance.**

49. Proportions for Gear-Teeth.—With gears of large size, and often with cast gears of all sizes, the circular-pitch system is used. In these cases, it is usual to have the addendum, whole depth, and thickness of the tooth conform to arbitrary rules based on the circular pitch.

The usual proportions are for cast gears: Addendum $= .3 \times$ circular pitch. Root $= .4 \times$ circular pitch. Thickness of tooth $= .48 \times$ circular pitch.

The gears most often met with are the cut gears of small and medium size like those, for example, on machine tools, which are almost invariably diametral-pitch gears. The teeth are cut from the solid with standard milling cutters, proportioned with the diametral pitch as a basis. This system is also coming into use for cast gearing. In all diametral-pitch gears, the addendum, in inches, is made equal to 1 divided by the diametral pitch, and the working depth to twice the addendum. The end clearance is usually taken equal to $\frac{1}{8}$ the addendum for cut gears, though The Brown & Sharpe Manufacturing Company use $\frac{1}{10}$ the thickness of the tooth on the pitch line as the clearance. The side clearance, or "backlash," is barely enough to give a good working fit, and seldom exceeds $\frac{1}{50}$ the pitch.

Using the above proportions, a 4-pitch gear will have the addendum $= 1 \div 4 = \frac{1}{4}$ inch; the working depth will be $2 \times \frac{1}{4} = \frac{1}{2}$ inch; and the clearance, if made $\frac{1}{8}$ the addendum, $\frac{1}{8} \times \frac{1}{4} = \frac{1}{32}$ inch. The whole length of the tooth will be $\frac{1}{2} + \frac{1}{32} = \frac{17}{32}$ inch. The thickness of the tooth will be one-half the circular pitch, nearly. In a 10-pitch wheel, the addendum will be $\frac{1}{10}$ inch and the length of the tooth $\frac{17}{80}$ inch; in a $2\frac{1}{2}$-pitch, it will be $1 \div 2\frac{1}{2} = \frac{2}{5}$ inch and the length $\frac{17}{20}$ inch.

FORMS OF GEAR-TEETH.

50. The forms of teeth used in ordinary practice form part of certain curves known as the *epicycloid*, *hypocycloid*, and *involute*.

51. The Epicycloidal Tooth.—In the so-called epicycloidal tooth, which more properly is called a *cycloidal* tooth, the face of the tooth is part of an epicycloid, and the flank part of a hypocycloid.

An **epicycloidal curve** is the path described by any point of a circle rolling, without slipping, on the outside of another circle. A **hypocycloid** is the path described by any point of a circle rolling, without slipping, on the inside of another circle.

Epicycloidal teeth can always be recognized by their appearance; they are formed by two curves that, commencing at the pitch circle, curve in opposite directions. Fig. 25 clearly exhibits the characteristic tooth form.

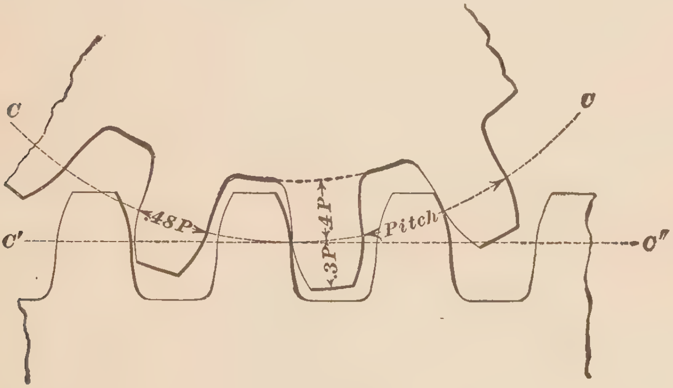

FIG. 25.

The use of epicycloidal teeth is being gradually abandoned, as they possess some practical defects, the chief defect being that the center-to-center distance of the two gears must be practically perfect in order to insure a uniform velocity of the driven gear.

52. Involute Teeth.—In Fig. 26 is shown the involute form of tooth, which is composed of but one curve.

The **involute** is the path described by any point of a string that is being wound on or off a cylinder, the cylinder being stationary. In the involute system, the sides of the teeth of the rack are straight lines, as shown in Fig. 26.

Involute teeth have two great advantages over epicycloidal teeth: (1) *They are stronger for the same pitch, as they are thicker at the root.* (2) *The gears may be spread* *slightly apart so that their pitch circles do not run tangent to*

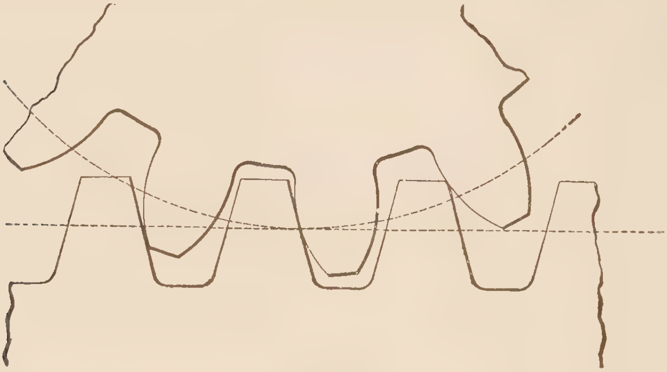

Fig. 26.

one another, without affecting the perfect action of the teeth to an appreciable extent.

GEAR CALCULATIONS.

53. The Circular-Pitch System.—For calculating the pitch diameter, number of teeth, etc. of gear-wheels, we have, for the circular-pitch system, the following rules,

where P = circular pitch in inches;
T = number of teeth;
D = pitch diameter of the gear in inches.

54. To find the pitch diameter of a gear-wheel in inches, when the pitch and number of teeth are given:

Rule 13.—*The pitch diameter equals the product of the pitch and the number of teeth divided by 3.1416.*

Or,
$$D = \frac{PT}{3.1416}.$$

Example.—What is the diameter of the pitch circle of a gear-wheel that has 75 teeth and whose pitch is 1.675 inches?

Solution.—Applying rule **13,** we have

$$D = \frac{1.675 \times 75}{3.1416} = 40 \text{ in. Ans.}$$

55. To find the number of teeth in a gear-wheel when the diameter and pitch are given:

Rule 14.—*The number of teeth equals the product of 3.1416 and the diameter divided by the pitch.*

Or, $$T = \frac{3.1416\,D}{P}.$$

Example.—The diameter of a gear-wheel is 40 inches and the pitch of the teeth is 1.675 inches; how many teeth are there in the wheel?

Solution.—Applying the rule just given, we have

$$T = \frac{3.1416 \times 40}{1.675} = 75 \text{ teeth.} \quad \text{Ans.}$$

56. To find the pitch of a gear-wheel when the diameter and the number of teeth are given:

Rule 15.—*The pitch of the teeth equals the product of 3.1416 and the diameter divided by the number of teeth.*

Or, $$P = \frac{3.1416\,D}{T}.$$

Example.—The diameter of a gear-wheel is 40 inches and it has 75 teeth; what is the pitch of the teeth?

Solution.—By rule **15**, we have

$$P = \frac{3.1416 \times 40}{75} = 1.675 \text{ in. pitch.} \quad \text{Ans.}$$

57. The Diametral-Pitch System.—The diameter of the gear-wheel, the number of teeth, etc. are given by the following rules, where $P_d =$ diametral pitch; $D_o =$ outside diameter; $N =$ number of teeth; and the other letters have the same meaning as in the three preceding rules.

58. To find the pitch diameter of the gear-wheel when the number of teeth and the pitch are given:

Rule 16.—*Divide the number of teeth by the diametral pitch.*

Or, $$D = \frac{N}{P_d}.$$

Example.—A wheel is to have 40 teeth, 4 pitch; what is its pitch diameter?

Solution.—By applying rule **16**, we have

$$D = \frac{40}{4} = 10 \text{ in.} \quad \text{Ans.}$$

59. To find the diameter over all, that is, the diameter of the blank from which the gear-wheel is cut, the number of teeth and the diametral pitch being given:

Rule 17.—*Add 2 to the number of teeth and divide by the diametral pitch.*

Or,
$$D_o = \frac{N + 2}{P_d}.$$

Example.—In the last example, what is the diameter over all the blank?

Solution.—Applying the rule just given, we get

$$D_o = \frac{40 + 2}{4} = 10\tfrac{1}{2} \text{ in.} \quad \text{Ans.}$$

60. The number of teeth and the outside diameter of the gear-wheel being known, to find the diametral pitch:

Rule 18.—*Add 2 to the number of the teeth and divide by the outside diameter.*

Or,
$$P_d = \frac{N + 2}{D_o}.$$

Example.—A gear-wheel has 60 teeth and is $6\frac{1}{5}$ inches in diameter over all; what is the diametral pitch?

Solution.—By applying rule **18**, we get

$$P_d = \frac{60 + 2}{6\frac{1}{5}} = 10. \quad \text{Ans.}$$

61. To find the diametral pitch, the number of teeth and the pitch diameter being known:

Rule 19.—*Divide the number of teeth by the pitch diameter.*

Or,
$$P_d = \frac{N}{D}.$$

Example.—A wheel has 90 teeth and its pitch diameter is 30 inches; what is the diametral pitch?

Solution.—By rule **19,** we get

$$P_a = \frac{90}{30} = 3. \quad \text{Ans.}$$

62. The pitch diameter and the diametral pitch being given, to obtain the number of teeth:

Rule 20.—*Multiply the pitch diameter by the diametral pitch.*

Or, $$N = D P_d.$$

Example.—How many teeth are there in a gear-wheel having a pitch diameter of 36 inches and a diametral pitch of 5?

Solution.—Applying the rule just given, we get

$$N = 36 \times 5 = 180 \text{ teeth.} \quad \text{Ans.}$$

63. The diameter over all and the diametral pitch being given, to find the number of teeth:

Rule 21.—*Subtract 2 from the product of the outside diameter and the diametral pitch.*

Or, $$N = D_o P_d - 2.$$

Example.—How many 10-pitch teeth has a gear-wheel having an outside diameter of $8\frac{1}{5}$ inches?

Solution.—By rule **21,** we get

$$N = 8\tfrac{1}{5} \times 10 - 2 = 80 \text{ teeth.} \quad \text{Ans.}$$

64. The standard diametral pitches used for cut gears are as follows: 2, $2\frac{1}{4}$, $2\frac{1}{2}$, $2\frac{3}{4}$, 3, $3\frac{1}{2}$, 4, 5, 6, 7, 8, 9, 10, 11, 12, 14, 16, 18, 20, 22, 24, 26, 28, 30, 32, 36, 40, 48. Gears having a diametral pitch differing from those given here are, usually, either very large or very small.

65. Diameters and Distances Between Centers.—The distance between the centers of two gear-wheels being known and the ratio of their speeds, the diameter of the pitch circle of the smaller wheel is given by the following rule,

where A = center-to-center distance;
R = revolutions per minute of large gear;
r = revolutions per minute of small gear;
d = pitch diameter of small gear.

Rule 22.—*Multiply twice the center-to-center distance by the number of revolutions of the large gear, and divide the product by the sum of the revolutions of the large and small gear.*

Or,
$$d = \frac{2\,A\,R}{R + r}.$$

Example.—Given, the distance between centers = 5½ inches; the small gear is to make 24 revolutions for every 8 revolutions of the large gear. What is the diameter of each gear?

Solution.—By rule **22**,

$$d = \frac{2 \times 5\frac{1}{2} \times 8}{8 + 24} = 2\tfrac{3}{4} \text{ in., the diameter of the small wheel.}$$

Then, as the center distance is equal to the sum of the radii of the two wheels, the radius of the large wheel is $5\frac{1}{2} - \frac{2\frac{3}{4}}{2} = 4\frac{1}{8}$ in., and its diameter is $4\frac{1}{8} \times 2 = 8\frac{1}{4}$ in. Ans.

66. Speed and Number of Teeth.—To calculate the number of teeth or the speed of one of two gear-wheels that are to gear together:

Let N = number of revolutions per minute of the driving wheel;
n = number of revolutions per minute of the driven wheel;
T = number of teeth in the driving wheel;
t = number of teeth in the driven wheel.

Rule 23.—*The number of teeth in the driving wheel equals the product of the number of teeth and number of revolutions of the driven wheel divided by the number of revolutions of the driving wheel.*

Or,
$$T = \frac{t\,n}{N}.$$

Example.—The driven wheel has 27 teeth and will make 66 revolutions per minute; if the driving wheel makes 99 revolutions per minute, how many teeth are there in the driving wheel?

Solution.—Applying rule **23**, we have

$$T = \frac{27 \times 66}{99} = 18 \text{ teeth. Ans.}$$

67. The number of revolutions per minute of the driving wheel and the driven wheel and the number of teeth in the driving wheel being given, to find the number of teeth in the driven wheel:

Rule 24.—*The number of teeth in the driven wheel equals the product of the number of teeth and revolutions per minute of the driving wheel divided by the number of revolutions per minute of the driven wheel.*

Or, $$t = \frac{TN}{n}.$$

EXAMPLE.—The driving wheel has 18 teeth and makes 99 revolutions per minute, and the driven wheel must make 66 revolutions per minute; how many teeth must there be in the driven wheel?

SOLUTION.—Applying the rule just given, we have

$$t = \frac{18 \times 99}{66} = 27 \text{ teeth. Ans.}$$

68. The number of teeth in the driving wheel and the driven wheel and the number of revolutions per minute of the driving wheel being given, to find the number of revolutions per minute of the driven wheel:

Rule 25.—*The number of revolutions per minute of the driven wheel equals the product of the number of teeth and number of revolutions of the driving wheel divided by the number of teeth of the driven wheel.*

Or, $$n = \frac{TN}{t}.$$

EXAMPLE.—There are 18 teeth in the driving wheel and it makes 99 revolutions per minute; how many revolutions per minute will the driven wheel make if it has 27 teeth?

SOLUTION.—Applying rule **25**, we have

$$n = \frac{18 \times 99}{27} = 66 \text{ R. P. M. Ans.}$$

69. The number of teeth in the driving wheel and the driven wheel and the number of revolutions per minute of the driven wheel being given, to find the number of revolutions per minute of the driving wheel:

Rule 26.—*The number of revolutions of the driving wheel equals the product of the number of teeth and revolutions of the driven wheel divided by the number of teeth of the driving wheel.*

Or, $$N = \frac{t\,n}{T}.$$

EXAMPLE 1.—If there are 27 teeth in the driven wheel and if it makes 66 revolutions per minute, how many revolutions per minute will the driving wheel make if it has 18 teeth?

SOLUTION.—Applying the rule just given, we have

$$N = \frac{27 \times 66}{18} = 99 \text{ R. P. M.} \quad \text{Ans.}$$

EXAMPLE 2.—In Fig. 27, the crank-shaft makes 60 revolutions per minute; the governor pulley is 4 inches in diameter; the bevel gear on the governor pulley shaft has 19 teeth; the bevel gear that meshes with it and drives the governor has 30 teeth. The governor is to make 95 revolutions per minute; what should be the size of the pulley on the crank-shaft?

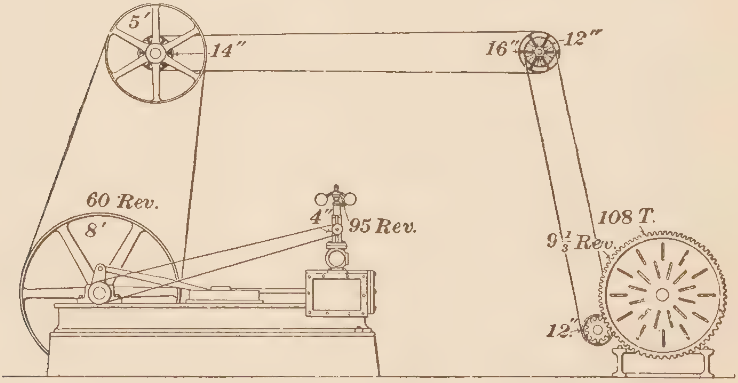

FIG. 27.

SOLUTION.—First determine the number of revolutions of the 4-inch pulley in order that the governor shall turn 95 times per minute. Applying rule **26**, $N = \frac{t\,n}{T} = \frac{30 \times 95}{19} = 150$ revolutions of gear on pulley shaft = revolutions of governor pulley. Now, applying rule **3**, the diameter of the pulley on the crank-shaft $= \frac{d\,n}{N} = \frac{4 \times 150}{60} = 10$ in.

Ans.

EXAMPLE 3.—In Fig. 27, the flywheel is 8 feet in diameter and drives a 5-foot pulley on the main shaft. A 14-inch pulley on the main shaft drives a 16-inch pulley on the countershaft. A 12-inch pulley on the countershaft drives a 12-inch pulley on a shaft on which is a pinion that meshes into a large gear attached to the face plate of a large lathe, and which has 108 teeth. How many teeth must the pinion have in order that the face plate may make $9\frac{1}{3}$ revolutions per minute?

SOLUTION.—Applying rule **5,** to find the revolutions per minute of the main shaft, $\frac{8 \times 60}{5} = 96$ R. P. M. Applying the same rule again to find the revolutions of the countershaft, $\frac{14 \times 96}{16} = 84$ R. P. M. Applying it once more to find the revolutions of the pulley that turns the small gear, $\frac{12 \times 84}{12} = 84$ R. P. M. Applying rule **23,** $\frac{108 \times 9\frac{1}{3}}{84} = 12$ teeth in pinion or driver. Ans.

EXAMPLES FOR PRACTICE.

1. The driving pulley makes 110 R. P. M. and is 21 inches in diameter; what should be the size of the driven pulley in order to make 385 R. P. M.? Ans. 6 in.

2. The main shaft of a certain shop makes 120 R. P. M. It is desired to have the countershaft make 150 R. P. M. There are on hand pulleys 16 inches, 24 inches, 28 inches, 35 inches, and 38 inches in diameter. Can two of these be used, or must a new pulley be ordered? Ans. Use the 28-inch and 35-inch pulleys.

3. The pinion (driver) makes 174 R. P. M. and the follower 24 R. P. M.; how many teeth must the pinion have if the follower has 87 teeth? Ans. 12 teeth.

4. If an engine flywheel is 66 inches in diameter and makes 160 R. P. M., what must be the diameter of the pulley on the main shaft to make 128 R. P. M.? Ans. 82½ in.

5. What is the pitch diameter of a gear whose circular pitch is 1¼ inches and has 28 teeth? Ans. 11.14 in.

6. How many teeth are there in a gear whose circular pitch is .7854 inch and which is 23 inches in diameter? Ans. 92 teeth.

7. What is the circular pitch of a gear whose diameter is 20.372 inches and which has 128 teeth? Ans. ½ in.

8. What is the pitch diameter of a gear-wheel having 80 teeth, 7 diametral pitch? Ans. $11\frac{3}{7}$ in.

9. What is the over-all diameter of a gear-wheel of 9 diametral pitch and 47 teeth? Ans. $5\frac{4}{9}$ in.

10. What is the diametral pitch of a gear-wheel having 90 teeth and an outside diameter of $5\frac{3}{4}$ inches? Ans. 16 pitch.

11. How many teeth of 20 diametral pitch can be cut in a gear-wheel having an outside diameter of $2\frac{1}{10}$ inches? Ans. 40 teeth.

12. If the center-to-center distance of two gear-wheels is 6 inches and the small gear is to make 4 revolutions for 1 revolution of the large gear, what must be the diameter of the large gear?
Ans. 9.6 in.

13. In a pair of gear-wheels, the driver has 48 teeth and the driven wheel 56 teeth. If the driven wheel makes 98 revolutions per minute, how many revolutions must the driver make? Ans. $114\frac{1}{3}$ R. P. M.

14. In a train of gears, the drivers have 16, 30, 24, and 18 teeth, respectively; the followers have 12, 24, 36, 40 teeth, respectively. If the first driver makes 80 R. P. M., how many R. P. M. will the last follower make? Ans. 40 R. P. M.

FIXED AND MOVABLE PULLEYS.

70. Pulleys are also used for hoisting or raising loads, in which case the frame that supports the axle of the pulley is called the **block.**

71. A **fixed pulley** is one whose block is not movable, as in Fig. 28. In this case, if the weight W be lifted by pulling down P, the other end of the cord W will evidently move the same distance upwards that P moves downwards; hence, P must equal W.

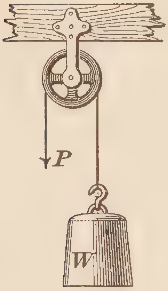

FIG. 28.

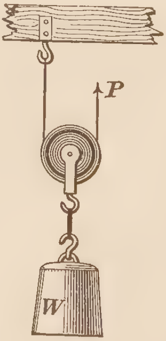

FIG. 29.

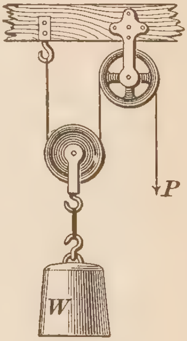

FIG. 30.

72. A **movable pulley** is one whose block is movable, as in Fig. 29. One end of the cord is fastened to the beam and the weight is suspended from the pulley, the other end of the cord being drawn up by the application of a force P. A little consideration will show that if P acts through a certain distance, say 1 foot, W will move through *half* that distance, or 6 inches; hence, a pull of 1 pound at P will lift 2 pounds at W.

The same would also be true if the free end of the cord were passed over a *fixed pulley*, as in Fig. 30, in which case the fixed pulley merely changes the direction in which P acts, so that a weight of 1 pound hung on the free end of the cord will balance 2 pounds hung from the *movable pulley*.

73. A **combination of pulleys,** as shown in Fig. 31, is sometimes used. In this case there are three movable and three fixed pulleys, and the amount of movement of W, owing to a certain movement of P, is readily found.

It will be noticed that there are 6 *parts* of the rope, not counting the free end; hence, if the movable block be lifted 1 foot, P remaining in the same position, there will be 1 foot of slack in each of the 6 parts of the rope, or 6 feet in all. Therefore, P must move 6 feet in order to take up this slack, or P moves 6 times as far as W. Hence, 1 pound at P will support 6 pounds at W, since the *force multiplied by the distance through which it moves equals the weight multiplied by the distance through which it moves.* It will also be noticed that there are three movable pulleys, and that $3 \times 2 = 6$.

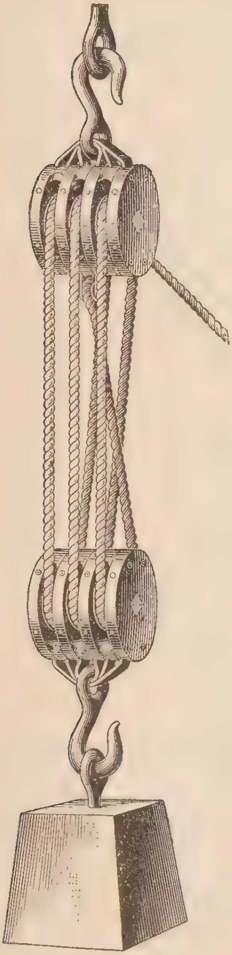

FIG. 31.

LAW OF COMBINATION OF PULLEYS.

74. Rule 27.—*In any combination of pulleys where one continuous rope is used, a load on the free end will balance a weight on the movable block as many times as great as itself as there are parts of the rope supporting the load, not counting the free end, assuming that there are no friction losses.*

The above law is good whether the pulleys are side by side, as in the ordinary block and tackle, or whether they are in line and beneath one another.

EXAMPLE.—In a block and tackle having 5 movable pulleys, how great a force must be applied to the free end of the rope to raise 1,250 pounds, assuming that there are no friction losses?

SOLUTION.—Since there are 5 movable pulleys, there must be 10 parts of the rope to support them. Hence, according to the above law, a force applied to the free end will support a load 10 times as great as itself, or the force $= \frac{1,250}{10} = 125$ lb. Ans.

75. Frictional Losses.—Owing to the friction of the sheaves and the friction and resistance to bending of the rope, it is not possible to move a weight with a block and tackle by applying the theoretical force calculated by the above rule. The actual pull required depends on such factors as the condition of the rope and the friction and number of the sheaves. By experiment it has been determined that under average conditions the load that can be raised with a block and tackle having 2 sheaves will average about 85 per cent. of the theoretical load; with 3 sheaves, about 80 per cent.; with 4 sheaves, about 75 per cent.; with 5 sheaves, about 70 per cent.; with 6 sheaves, about 66 per cent.; with 7 sheaves, about 63 per cent.; and with 8 sheaves, about 60 per cent. No records of tests with a larger number of sheaves have been made public; the decrease in the load that can be lifted will probably vary in about the same ratio.

Let f_a = the force applied at the free end of the rope;
c = the percentage of the actual load to the theoretical load, expressed decimally,
m = the number of parts of rope supporting the load;
l = the load to be raised.

Then, the force that is actually required to raise a given load, may be found approximately by

Rule 28.—*Divide the load by the product of the number of parts of the rope supporting the load and the per cent. corresponding to the number of sheaves used.*

Or, $$f_a = \frac{l}{mc}.$$

Example.—In an ordinary block and tackle having six parts of the rope supporting the load, as shown in Fig. 31, how great a force must actually be applied to raise 2,000 pounds?

Solution.—According to Art. **75,** c may be taken as .66, there being 6 parts of the rope. Applying rule **28,**

$$f_a = \frac{2{,}000}{6 \times .66} = 505 \text{ lb., nearly.} \quad \text{Ans.}$$

76. The load that may actually be raised with a block and tackle will be approximately given by the following rule:

Rule 29.—*Multiply the force to be applied to the free end of the rope by the number of parts of rope supporting the load and by the per cent. corresponding to the number of sheaves used.*

Or, $$l = f_a mc.$$

Example.—If there are enough men pulling on the free end of the rope of a block and tackle having four parts of the rope supporting the load to exert a force of 250 pounds, what weight may they expect to raise?

Solution.—There being 4 parts of the rope, by Art. **75,** $c = .75$. Then by rule **29,**

$$l = 250 \times 4 \times .75 = 750 \text{ lb.} \quad \text{Ans.}$$

77. If the free end of the hauling rope passes first around a stationary sheave, as in Fig. 31, it does not make any difference in what direction in the plane of the sheave the rope is pulled. If it passes first around a movable sheave, however, as in Fig. 29, the pull must be exerted in a line parallel to the line of action of the resistance, or a line joining the centers of the movable and stationary sheaves, in order to obtain the maximum effect. If the rope pulls on the movable sheave at an angle, its effect will be a displacement sideways, instead of straight up.

78. The **Weston differential pulley block,** more commonly known as a **chain hoist,** is shown in Fig. 32. With the help of this device, one man can hoist a load far beyond what can be done with the ordinary block and tackle. There are two pulleys slightly different in size alongside of each other in the stationary block and rigidly connected so as to turn together. An endless chain passes over both pulleys and supports the movable block in one of its bights. The other loop, or bight, is free and forms the hauling part. This kind of a pulley block possesses the valuable property that the load can be stopped anywhere by simply ceasing to haul on the hoisting part of the free loop.

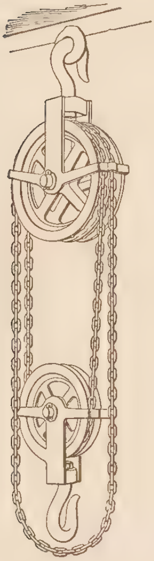
FIG. 32.

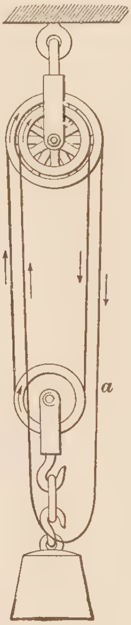

FIG. 33.

In Fig. 33 the differential pulley block is shown in diagrammatic form. Suppose that the part *a* of the free bight is pulled downwards, that is, in the direction of the arrow. Evidently, the upper pulleys will both turn in the direction of the hands of a watch, as indicated by the curved arrow. Then, the left-hand part of the bight supporting the movable pulley will move upwards, and the right-hand part downwards. But as the large and small pulley of the upper block move together, it follows that the left-hand side of the bight supporting the movable sheave will run faster over the large pulley than the right-hand side of the bight will run downwards over the small pulley. The result is that the upward motion of the movable block will be equal to one-half the difference of the distances passed over in the same time by fixed points in both sides of the bight.

79. **Differential chain hoists** are invariably so constructed that one man can hoist any load up to the maximum capacity of the hoists. This maximum load will usually be found stamped on the upper block, and there is, hence, no need in practice of calculating the force that must be exerted to hoist a load with a chain hoist.

EXAMPLES FOR PRACTICE.

1. In a block and tackle with four parts of the rope supporting the load, (*a*) what force will be required to lift 300 pounds, assuming that there are no friction losses? (*b*) What force may be expected to be actually required, taking friction into account?

Ans. { (*a*) 75 lb.
(*b*) 100 lb.

2. A man weighing 120 pounds throws his whole weight on the free end of the rope of a block and tackle with six parts of the rope supporting the load. What weight can he raise, (*a*) assuming that there are no friction losses? (*b*) taking friction into account?

Ans. { (*a*) 720 lb.
(*b*) 475 lb., about

THE INCLINED PLANE.

80. An **inclined plane** is a slope, or flat surface, making an angle with another surface or base plane.

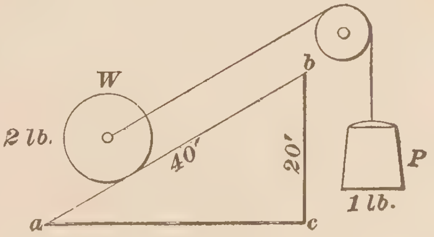

FIG. 34.

81. Three cases may arise in practice with an inclined plane having a horizontal base plane: (1) Where the force acts parallel to the plane, as in Fig. 34. (2) Where the force acts parallel to the base, as in Fig. 35. (3) Where the force acts at an angle to the plane or to the base, as in Fig. 36.

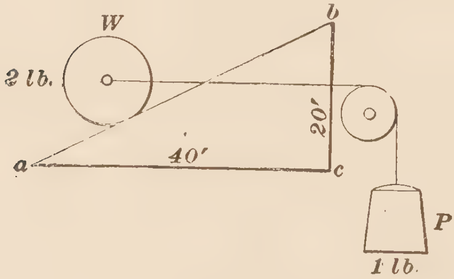

FIG. 35.

82. Relation Between Force and Weight.—In Fig. 34, the relation existing between the force and the weight is easily found. The weight ascends a distance equal to cb, or the height of the inclined plane, while the force acts through a distance equal to ab, or the length of the inclined plane. Therefore,

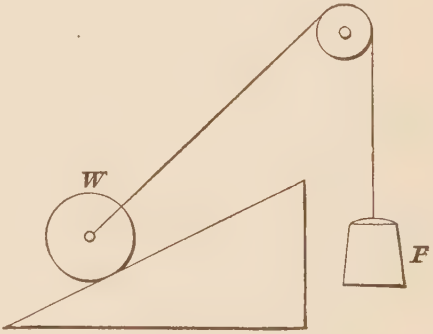

FIG. 36.

Rule 30.—*To find the force, multiply the weight by the height of the plane and divide the product by the length of the plane. To find the weight that can be raised, multiply the force by the length of the plane and divide this product by the height of the plane.*

Or, let
P = force;
l = length of the plane;
b = base of the plane;
h = height of plane;
W = weight.

Then, $P=\frac{Wh}{l}$ and $W=\frac{Pl}{h}$.

EXAMPLE 1.—The length of an inclined plane is 40 feet and its height is 5 feet; what force is required to sustain a weight of 100 pounds?

Solution.—Applying rule **30**, we have

$$P = \frac{100 \times 5}{40} = 12\tfrac{1}{2} \text{ lb.} \quad \text{Ans.}$$

Example 2.—The length of an inclined plane is 8 feet and its height is 15 inches; what weight will a force of 120 pounds support?

Solution.—Applying the rule just given,

$$W = \frac{120 \times 96}{15} = 768 \text{ lb.} \quad \text{Ans.}$$

83. In Fig. 35, the force is supposed to act parallel to the base for any position of W; therefore, while W is moving from the level $a\,c$ to b, or through the height $c\,b$ of the inclined plane, P will move through a distance equal to the length of the base $a\,c$. When the force acts parallel to the base, we have

Rule 31.—*To find the force required, multiply the weight by the height of the plane and divide the product by the length of the base of the plane. To find the weight that can be raised, multiply the force by the length of the base of the plane and divide the product by the height of the plane.*

Or,
$$P = \frac{W\,h}{b} \text{ and } W = \frac{P\,b}{h}.$$

Example 1.—With a base 30 feet long and a height of 6 feet, what force will sustain a weight of 75 pounds?

Solution.—By rule **31**,

$$P = \frac{75 \times 6}{30} = 15 \text{ lb.} \quad \text{Ans.}$$

Example 2.—A force of 12 pounds sustains a weight on a plane whose base is 6 feet long and whose height is 18 inches; find the weight.

Solution.—Applying the rule just given, we have

$$W = \frac{12 \times 6}{1\tfrac{1}{2}} = 48 \text{ lb.} \quad \text{Ans.}$$

For Fig. 36, no rule can be given. The ratio of the power varies as the position of W varies; it may be determined by the principle of the resolution of forces. As W

shifts its position, the angle that its cord makes with the face of the plane varies, and the magnitude of the force P depends on this angle; the smaller the angle the less is P required to be in order to support a given weight on a given plane.

84. Applications of the Inclined Plane.—The wedge is a movable inclined plane that in various modifications serves for many different purposes. In the form of a stake wedge, as shown in Fig. 37, it serves to move heavy weights or to separate closely joined pieces of material. In the form of a cold chisel or knife, it serves to sever substances. In the form of a key, it serves to move the brasses in the different pin-joint connections of a steam engine. Familiar examples of inclined planes about machinery are the keys used for fastening a crank to the crankshaft or a pulley to the shaft; in some designs of crossheads, the cotters securing the crosshead to the piston rod are an example of an inclined plane; as are also the adjustable shoes in many designs of crossheads for steam engines.

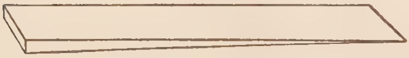

FIG. 37.

85. Taper of Keys.—In keys, cotters, and stake wedges, it is customary to express the inclinations as a certain amount per foot, or taper per foot. Thus, in Fig. 38, the inclination would be 3 − 2 = 1 inch in 12 inches, and the taper would be, as there are 12 inches in a foot, 1 inch per foot. The taper per foot can be ascertained by the following rule,

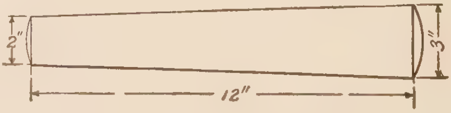

FIG. 38.

where a = the large diameter in inches;
b = the small diameter in inches;
l = the length in inches;
T = taper in inches per foot.

Rule 32.—*Subtract the small diameter from the large diameter, multiply the remainder by 12, and divide the product by the length of the key.*

Or,
$$T = \frac{(a - b)\,12}{l}.$$

EXAMPLE.—A key measures 4 inches at the large end and 3 inches at the small end; if its length is 16 inches, what is the taper per foot?

SOLUTION.—Applying the rule just given, we get

$$T = \frac{(4 - 3) \times 12}{16} = \tfrac{3}{4} \text{ in. per ft.} \quad \text{Ans.}$$

86. Where tapered keys are used for adjusting brasses, it is often desirable to know how much the brasses will be closed in on the pin by a given movement of the key, or how much to back out a key that has been driven home until the brasses nip the pin, the clearance between the brasses and the pin necessary for satisfactory running having been decided on.

Let T = taper in inches per foot;
d = distance the key is driven in inches;
x = amount the brass is moved in inches.

Rule 33.—*To find the amount the brass has been moved, multiply the taper by the distance the key is driven and divide the product by 12. To find how far to back out a key, multiply the amount the brass is to be moved back by 12 and divide by the taper.*

Or,
$$x = \frac{T d}{12} \text{ and } d = \frac{12\,x}{T}.$$

EXAMPLE.—In Fig. 39, the key has been driven home until the brasses nip the pin; it has been determined to run the brasses with a clearance of $\frac{1}{64}$ inch; how much must the key be backed out, its taper being $\frac{3}{4}$ inch per foot?

SOLUTION.—By rule **33**,

$$d = \frac{12 \times \frac{1}{64}}{\frac{3}{4}} = \tfrac{1}{4} \text{ in.} \quad \text{Ans.}$$

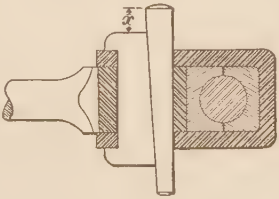

FIG. 39.

A ready way of measuring the distance the key is driven back is to measure from the top surface of the cotter to the upper edge of the key while the key is home. Add the amount the key is to be driven back, and move it until a rule shows it to have been driven the right amount. This method is preferable to making scriber marks on the flat sides of the key, as there is no liability of getting mixed up by the multiplicity of marks sure to be found after years of service.

87. Keys for fastening pulleys to shafts in order to cause them to rotate together belong usually to one of the three classes shown in Fig. 40. The one shown at *a* is a concave key that is hollowed out to fit the shaft. As it holds the pulley by friction alone, it is only suitable for light work. In case such a key slips, its holding power can be increased a little by hollowing out its concave side to a radius smaller than the radius of the shaft. The flat key is shown in Fig. 40 at *b*; a flat surface is cut on the shaft, and, consequently, the key is more effective than the concave key, as far as slipping under a rotative strain is concerned. The sunk key shown in Fig. 40 at *c* is the most effective, since it is impossible for the pulley to turn on the shaft without shearing off the key. Keys for fastening pulleys, etc. are usually given a taper of from ⅛ to ¼ inch per foot.

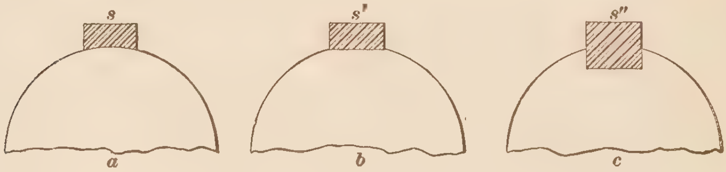

FIG. 40.

EXAMPLES FOR PRACTICE.

1. An inclined plane is 30 feet long and 7 feet high; what force is required to roll a barrel of flour weighing 196 pounds up the plane, the friction being neglected? Ans. 45.7 lb.

2. What is the taper per foot of a stake wedge measuring $\frac{1}{32}$ inch at the small end and $\frac{1}{4}$ inch at the large end, its length being 4 inches?

Ans. $\frac{21}{32}$ in. per ft.

3. Given, a key having a taper of 1 inch per foot. How much must it be backed out to give the brasses a working clearance of $\frac{1}{32}$ inch?

Ans. $\frac{3}{8}$ in.

THE SCREW.

88. A **screw** is a cylinder having a helical projection winding around its circumference. This helix is called the **thread** of the screw. The distance that a point on the helix is drawn back or advanced in the direction of the length of the screw during one turn is called the **pitch** of the screw. The screw in Fig. 41 is turned in a nut *a* by means of a force applied at the end of the handle *P*. For one complete revolution of the handle, the screw will be advanced lengthwise an amount equal to the pitch. If the nut is fixed and a weight placed upon the end of the screw, as shown, it will be raised vertically, by one revolution of the screw, a distance equal to the pitch. During this revolution, the force at *P* will move through a distance equal to the circumference of a circle whose radius is *P F*. Therefore, *W × pitch of thread = P × circumference of circle through which P moves.*

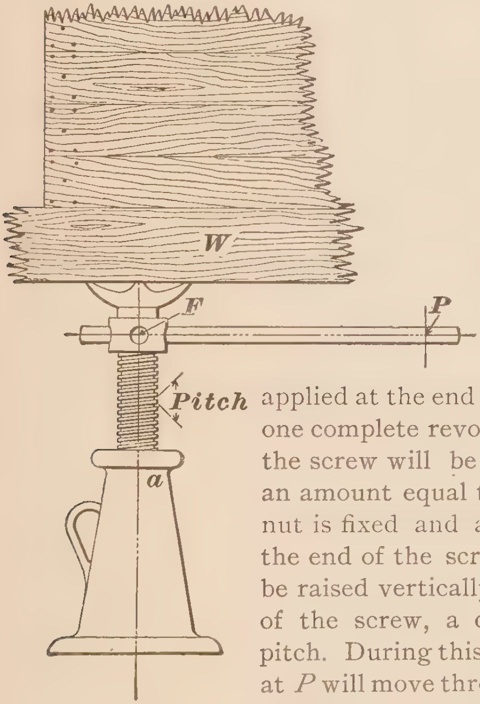

FIG. 41.

89. Owing to the friction between the nut and the screw and between the pivot of the screw and its cap, the weight that can be raised with a screw, or the force that can be exerted by it in the direction of its axis, is far less than that calculated on the assumption that friction does not exist. The friction depends on the pitch, the condition of the thread on the screw and in the nut, and the nature of the lubricant.

From a long series of experiments, Mr. Wilfred Lewis has deduced a formula for determining a factor by which to multiply the weight that could be raised if there were no friction, so as to determine the probable actual weight that can be raised.

Let d = diameter of screw in inches;
p = pitch in inches;
E = the number by which to multiply the theoretical weight in order to obtain the approximate actual weight.

Rule 34.—*Add the diameter of the screw to the pitch and divide the pitch by this sum; the quotient so obtained will be the factor by which to multiply the theoretical weight in order to determine the approximate actual weight.*

Or,
$$E = \frac{p}{p + d}.$$

Example.—With a screw having 4 threads to the inch and 2 inches in diameter, what is the factor by which to determine how much of the theoretical load can probably be raised?

Solution.—Since there are 4 threads per inch, the pitch is 1 ÷ 4, or $\frac{1}{4}$ inch. Then applying rule **34**, we get

$$E = \frac{\frac{1}{4}}{\frac{1}{4} + 2} = .11. \quad \text{Ans.}$$

90. The weight that can be lifted when friction is neglected, or the force that can be exerted by a screw when a given force is exerted on the lever, is given by the following rule,

where W = weight or force exerted by the screw;
P = force applied to the end of the lever;
p = pitch;
r = distance from the axis of the screw to the point of application of the force P.

Rule 35.—*Multiply the force applied to the lever, in pounds, by the distance of its point of application from the axis of the screw, in inches, and multiply this product by 6.2832; divide the result by the pitch in inches and the quotient will be the theoretical force that will be exerted by the screw.*

Or, $$W = \frac{6.2832\, P\, r}{p}.$$

91. To find the theoretical force that must be applied to the lever to raise a given weight or to exert a given force:

Rule 36.—*Multiply the weight or given force by the pitch and divide this product by the product of 6.2832 and the distance from the axis of the screw to the point on the lever where the force is applied.*

Or, $$P = \frac{W p}{6.2832\, r}.$$

92. To obtain the probable actual weight that may be raised, multiply the result obtained by rule **35** by the value obtained from rule **34.** To obtain the probable force that must be applied to the lever in order to lift a given weight, divide the result obtained from rule **36** by the value obtained from rule **34.**

Example 1.—A screw jack has a screw $1\frac{3}{4}$ inches diameter with 5 threads to the inch. If a man pulls at the end of the lever 14 inches from the axis with a force of 40 pounds, what weight may he expect to raise?

Solution.— 5 threads per inch = $\frac{1}{5}$-inch pitch.

By rule **35**, $$W = \frac{6.2832 \times 40 \times 14}{\frac{1}{5}} = 17{,}593 \text{ pounds, nearly.}$$

By rule **34**, $$E = \frac{\frac{1}{5}}{\frac{1}{5} + 1\frac{3}{4}} = .1, \text{ nearly.}$$

Then, as just stated, the probable weight that may be raised is

$$17{,}593 \times .1 = 1{,}759.3 \text{ lb.} \quad \text{Ans.}$$

Example 2.—The pull that an ordinary man can exert at the end of a lever is usually taken at 40 pounds; would it be possible for one man to raise a weight of 3 tons with a jack-screw having 4 threads to the inch and 2½ inches in diameter when the longest lever available is 24 inches?

Solution.—According to rule **36,** the theoretical force that must be exerted is

$$\frac{3 \times 2000 \times \frac{1}{4}}{6.2832 \times 24} = 9.9 \text{ pounds, nearly.}$$

By rule **34,**

$$E = \frac{\frac{1}{4}}{\frac{1}{4} + 2\frac{1}{2}} = .091, \text{ nearly.}$$

Then, as just stated, the probable actual force required is 9.9. 9.9 ÷ .091 = 109 pounds, about, or more than a single man may be expected to exert steadily. Ans.

SCREWS USED AS FASTENINGS.

93. In machine construction, screws are used more frequently as fastening devices than as a means of transmitting motion or producing pressure. Screws used as fastening devices may be divided into two general classes, which are distinguishable from each other by the form of thread used: (1) *Metal screws*, or screws intended to be screwed into metals. (2) *Wood screws*, or screws intended for wood.

94. Metal screws have either a sharp **V** thread or the United States standard thread shown in Fig. 42. In the latter, as also in the sharp **V** thread, the sides of the thread form a 60° angle. In the United States standard thread, the thread is flattened at the top and bottom, as shown in

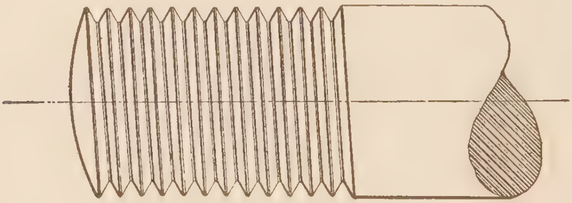

Fig. 42.

the figure. The advantage of this is that it makes a stronger screw for equal pitches and diameter.

95. Metal screws are divided by the trade into *machine bolts*, *capscrews*, and *machine screws*.

96. In **machine bolts,** or **bolts,** for short, one of which is shown in Fig. 43 (*a*), the shank and head are left rough, or just as they come from the bolt-heading machine. Machine bolts are regularly made in lengths varying by half inches from $1\frac{1}{2}$ to 8 inches. Above 8 inches and up to 20 inches the length varies by whole inches. The length of a machine bolt is always measured under the head, as the distance l in Fig. 43 (*a*).

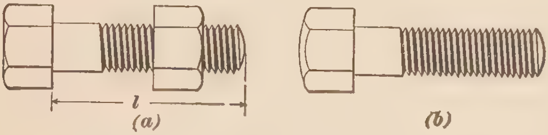

FIG. 43.

97. The standard diameters in which machine bolts are regularly made are $\frac{1}{4}$, $\frac{5}{16}$, $\frac{3}{8}$, $\frac{7}{16}$, $\frac{1}{2}$, $\frac{9}{16}$, $\frac{5}{8}$, $\frac{3}{4}$, $\frac{7}{8}$, and 1 inch. Sizes differing from these are special and can only be obtained on special order. By the *diameter of a bolt* is always meant its diameter over the top of the thread. Machine bolts can be obtained with either hexagonal or square heads and with hexagonal or square nuts.

98. **Capscrews,** one of which is shown in Fig. 43 (*b*), are bolts with either hexagonal or square heads; they have the shanks turned and the heads finished all over by milling and turning. Unlike machine bolts, they are usually furnished without nuts, unless especially ordered.

99. Machine bolts and capscrews are made with a pitch of thread corresponding to the United States standard thread, as shown in Table II.

TABLE II.

STANDARD THREADS PER INCH.

Diameter. Inches.	Threads per Inch.	Diameter. Inches.	Threads per Inch.
$\frac{1}{4}$	20	$1\frac{1}{4}$	7
$\frac{5}{16}$	18	$1\frac{3}{8}$	6
$\frac{3}{8}$	16	$1\frac{1}{2}$	6
$\frac{7}{16}$	14	$1\frac{5}{8}$	$5\frac{1}{2}$
$\frac{1}{2}$	13	$1\frac{3}{4}$	5
$\frac{9}{16}$	12	$1\frac{7}{8}$	5
$\frac{5}{8}$	11	2	$4\frac{1}{2}$
$\frac{3}{4}$	10	$2\frac{1}{4}$	$4\frac{1}{2}$
$\frac{7}{8}$	9	$2\frac{1}{2}$	4
1	8	$2\frac{3}{4}$	4
$1\frac{1}{8}$	7	3	$3\frac{1}{2}$

The only deviation from this table of threads is in $\frac{1}{2}$-inch square and hexagonal capscrews, where the usual number of threads for a sharp **V** thread is 12 per inch instead of 13, as called for. Capscrews are usually furnished with a sharp **V** thread, but can now be obtained with the United States standard thread. Machine bolts usually have the United States standard thread and the number of threads per inch called for in the table.

100. Machine screws are small screws with a slotted head. They are known to the trade as *flat heads*, shown in Fig. 44 (*a*); *round heads*, shown in Fig. 44 (*b*); *button heads*, shown in Fig. 44 (*c*), and *fillister heads*, shown in Fig. 44 (*d*).

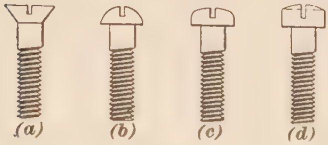

FIG. 44.

101. Screws for uniting metals are made in a variety of forms that differ from those here shown to suit special conditions. Some of these are setscrews, carriage bolts, tire bolts, boiler patch bolts, elevator bolts, etc. Most of these can only be obtained on special order, although some of the more common sizes may be found in stock.

102. Wood screws are either flat headed or round headed, with a slotted head. When wood screws have a square head for a wrench, instead of a slot for a screwdriver, they are known as *lagscrews*, or *coach screws.*

VELOCITY RATIO AND EFFICIENCY.

VELOCITY RATIO.

103. By the term **velocity ratio** is meant the ratio of the distance through which the force acts to the corresponding distance the weight moves. Thus, if the force acts through a distance of 12 inches while the weight moves 1 inch, the velocity ratio is 12 to 1, or 12; that is, P moves 12 times as fast as W, P representing the force and W the weight.

If the velocity ratio is known, the theoretical weight, that is, the weight that any machine will raise under the assumption that there are no frictional resistances, can be found by multiplying the force by the velocity ratio. If the velocity ratio is 8.7 to 1, or 8.7, $W = 8.7 \times P$, since $W \times 1 = P \times 8.7$.

104. From the preceding article it is seen that when the theoretical weight which may be lifted by a given force is known, the velocity ratio can be obtained by dividing this weight by the force. For example, if a force of 1 pound is applied to the handle of a screw whose pitch is $\frac{1}{4}$ inch, at a distance of 14 inches from the axis, it will lift, according to rule **35**, a theoretical weight equal to

$$W = \frac{6.2832 \times 1 \times 14}{\frac{1}{4}} = 351.86 \text{ pounds.}$$

Since a force of 1 pound lifts a theoretical weight of 351.86 pounds, the velocity ratio = 351.86 ÷ 1 = 351.86. This result can be verified by dividing the distance the point at which the force is applied moves in one revolution by the distance the screw advances in the same time. The point at which the force is applied travels in one revolution $2 \times 14 \times 3.1416 = 87.965$ inches. The screw advances in one revolution a distance equal to its pitch, or $\frac{1}{4}$ inch; hence, the velocity ratio = $87.965 \div \frac{1}{4} = 351.86$, the same value as obtained before.

EFFICIENCY.

105. The amount of work that can be obtained from a machine is always less than that required to operate it. This is due to the fact that some work is always lost in overcoming friction or other resistances. The ratio of the amount of useful work which may be obtained from a machine to the corresponding amount of work required to drive it is called the **efficiency** of the machine. In the case of the simple machines, previously described, the efficiency is the quotient found by dividing the weight which can be actually raised by that which may be raised theoretically.

Example.—It is found that with a certain block and tackle having two movable sheaves a force of 50 pounds will lift a weight of 172 pounds; what is the efficiency?

Solution.—According to rule **27,** the theoretical weight that can be raised, there being 4 parts of the rope supporting the load, is $4 \times 50 = 200$ lb. Then,

Efficiency $= \frac{172}{200} = .86$, or 86 per cent. Ans.

106. If the efficiency of a machine is known, the actual force required to raise a given load may be found by dividing the theoretical force required to raise the load by the efficiency. Thus, if it is found that a theoretical force of 85 pounds is required to lift a certain load by means of a screw, and the efficiency is 40 per cent., the actual force needed is $85 \div .40 = 212.5$ pounds.

107. If the efficiency is known, the weight that a certain force will raise may be found by multiplying together the force, velocity ratio, and the efficiency. Thus, if a certain machine has a velocity ratio of 6.5 and an efficiency of 78 per cent., a force of 140 pounds will raise a weight of $140 \times 6.5 \times .78 = 709.9$ pounds.

EXAMPLES FOR PRACTICE.

1. The distance from the axis of a screw to the point on the handle where the force is applied is 12 inches. The screw has 8 threads per inch and is 1 inch in diameter. (*a*) What force is necessary to raise a weight of 1,248 pounds, assuming that there is no friction? (*b*) What force will probably be actually required?

Ans. $\left\{\begin{array}{ll}(a) & 2.07\text{ lb., nearly.}\\ (b) & 19\text{ lb., nearly.}\end{array}\right.$

2. What weight can actually be raised with a screw jack having a screw 1¼ inches in diameter and 6 threads per inch, when a man is pulling with a force of 40 pounds at the end of a bar 16 inches long?

Ans. 2,838 lb., nearly.

3. In example 2, what is the velocity ratio? Ans. 603, nearly.

4. In example 2, what is the efficiency? Ans. 11¾ per cent.

MECHANICS OF FLUIDS.

HYDROSTATICS.

1. **Hydrostatics** treats of liquids at rest under the action of forces.

2. Liquids are very nearly *incompressible.* A pressure of 15 pounds per square inch compresses water less than $\frac{1}{20000}$ of its volume.

3. **Hydrostatic Pressure.**—Fig. 1 represents two cylindrical vessels of exactly the same size. The vessel *a* is fitted with a wooden block of the same size as, and free to move in, the cylinder; the vessel *b* is filled with water, whose depth is the same as the length of the wooden block in *a*. Both vessels are fitted with air-tight pistons *P*, each of whose areas are 10 square inches.

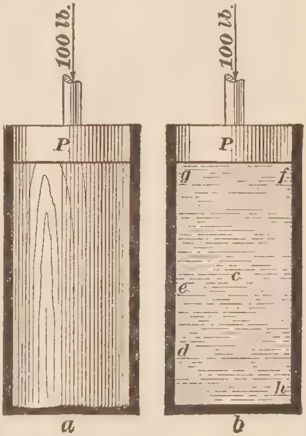

FIG. 1.

Suppose, for convenience, that the weights of the pistons, block, and water be neglected, and that a force of 100 pounds be applied to both pistons. The pressure per square inch will be $\frac{100}{10} = 10$ pounds. In the vessel *a* this pressure will be transmitted to the bottom of the vessel, and will be 10 pounds per square inch; it is easy to see that there will

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be no pressure on the sides. In the vessel *b* an entirely different result is obtained. The pressure on the bottom will be the same as in the other case, that is, 10 pounds per square inch, but, owing to the fact that the molecules of the water are perfectly free to move, this pressure of 10 pounds per square inch is *transmitted in every direction with the same intensity;* that is to say, the pressure at any point *c*, *d*, *e*, *f*, *g*, *h*, etc., due to the force of 100 pounds, is exactly the same, and equals 10 pounds per square inch.

4. The foregoing fact may be easily proved experimentally by means of an apparatus like that shown in Fig. 2. Let the area of the piston *a* be 20 square inches; of *b*, 7 square inches; of *c*, 1 square inch; of *d*, 6 square inches; of *e*, 8 square inches; and of *f*, 4 square inches.

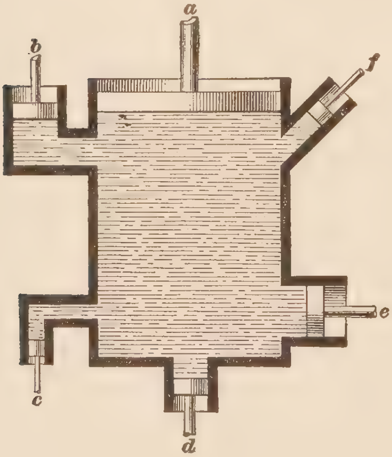


FIG. 2.

If the pressure due to the weight of the water be neglected, and a force of 5 pounds be applied at *c* (whose area is 1 square inch), a pressure of 5 pounds per square inch will be transmitted in all directions, and in order that there shall be no movement, a force of 6 × 5 = 30 pounds must be applied at *d*, 40 pounds at *e*, 20 pounds at *f*, 100 pounds at *a*, and 35 pounds at *b*.

If a force of 99 pounds were applied to *a*, instead of 100 pounds, the piston *a* would rise, and the other pistons *b*, *c*, *d*, *e*, and *f* would move inwards; but, if the force applied to *a* were 100 pounds, they would all be in equilibrium. If 101 pounds were applied at *a*, the pressure per square inch would be $\frac{101}{20} = 5.05$ pounds, which would be transmitted in all directions; and, since the pressure due to the load on *c* is only 5 pounds per square inch, it is now evident that the piston *a* will move downwards, and the pistons *b*, *c*, *d*, *e*, and *f* will be forced outwards.

5. Pascal's Law.—The whole of the preceding may be summed up as follows:

The pressure per unit of area exerted anywhere on a liquid is transmitted undiminished in all directions and acts with the same force on all surfaces, in a direction at **right angles** *to those surfaces.*

This law, first discovered by Pascal, and accordingly named after him, is the most important one in **hydrostatics.** Its meaning should be thoroughly understood.

EXAMPLE.—If the area of the piston *e*, in Fig. 2, were 8.25 square inches and a force of 150 pounds were applied to it, what forces would have to be applied to the other pistons to keep the water in equilibrium, assuming that their areas were the same as given before?

SOLUTION.— $\frac{150}{8.25} = 18.182$ pounds per square inch, nearly.

$$\left.\begin{array}{rl} 20 \times 18.182 = 363.64 \text{ lb.} &= \text{force to balance } a. \\ 7 \times 18.182 = 127.274 \text{ lb.} &= \text{force to balance } b. \\ 1 \times 18.182 = 18.182 \text{ lb.} &= \text{force to balance } c. \\ 6 \times 18.182 = 109.092 \text{ lb.} &= \text{force to balance } d. \\ 4 \times 18.182 = 72.728 \text{ lb.} &= \text{force to balance } f. \end{array}\right\} \text{Ans.}$$

6. *The pressure due to the weight of a liquid may be downwards, upwards, or sidewise.*

7. Downward Pressure.—In Fig. 3 the pressure on the bottom of the vessel *a* is, of course, equal to the weight of the water it contains. If the area of the bottom of the vessel *b* and the depth of the liquid contained in it are the same as in the vessel *a*, the pressure on the bottom of *b* will be the same as on the bottom of *a*. Suppose the bottoms of the vessels *a* and *b* are 6 inches square, and that the part *c d* in the vessel *b* is 2 inches square, and that they are filled with water. Then, since 1 cubic foot of water weighs 62.5 pounds, the weight of 1 cubic inch of water is $\frac{62.5}{1,728}$ pound = .03617 pound. The number of cubic inches in $a = 6 \times 6 \times 24 = 864$ cubic inches; and the weight of the water is $864 \times .03617 = 31.25$ pounds. Hence, the total pressure on the bottom of the vessel *a* is 31.25 pounds, or $\frac{31.25}{6 \times 6} = .868$ pound per square inch.

The pressure in *b* due to the weight contained in the part *b c* is $6 \times 6 \times 10 \times .03617 = 13.02$ pounds.

The weight of the part contained in *c d* is $2 \times 2 \times 14 \times .03617 = 2.0255$ pounds, and the weight per square inch of area in *c d* is $\frac{2.0255}{4} = .5064$ pound.

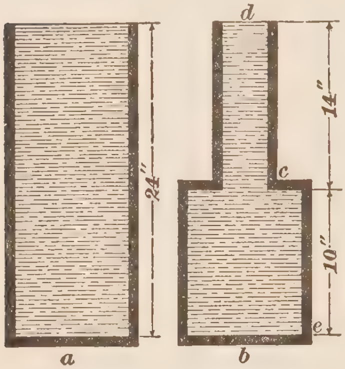


FIG. 3.

8. According to Pascal's law, this weight (pressure) is transmitted equally in all directions, therefore, an extra weight of .5064 pound is imposed on every square inch of the bottom of *b c*; the area of this is $6 \times 6 = 36$ square inches, and the pressure on it due to the water contained in *c d* is, therefore, $36 \times .5064 = 18.23$ pounds; thus, we have a total pressure on the bottom of vessel *b* of $13.02 + 18.23 = 31.25$ pounds, the same as in vessel *a*. As a result of the above law, there is also an upward pressure of .5064 pound acting on every square inch of the *top* of the enlarged part *b c*.

If an additional pressure of 10 pounds per square inch were applied to the upper surface of both vessels, the total pressure on each bottom would be $31.25 + (6 \times 6 \times 10) = 31.25 + 360 = 391.25$ pounds.

If this pressure were to be obtained by means of a weight placed on each piston (as shown in Fig. 1 at *a* and *b*), we would have to put a weight of $6 \times 6 \times 10 = 360$ pounds on the piston in vessel *a*, Fig. 3, and one of $2 \times 2 \times 10 = 40$ pounds on the piston in vessel *b*.

9. The general law for the downward pressure on the bottom of any vessel:

Rule 1.—*The pressure on the bottom of a vessel containing a fluid is independent of the shape of the vessel, and is equal to the weight of a column of the fluid, the area of whose base is equal to that of the bottom of the vessel and whose altitude is the distance between the bottom and the upper surface of the fluid, increased by the pressure per unit of area on the upper surface of the fluid multiplied by the area of the bottom of the vessel, in case there is any pressure on the surface.*

10. Suppose that the vessel *b*, in Fig. 3, were inverted, as shown in Fig. 4, the pressure on the bottom would still be .868 pound per square inch, but it would require a weight of 3,490 pounds on a piston at the upper surface to make the pressure on the bottom 391.25 pounds, instead of a weight of 40 pounds, as in the other case.

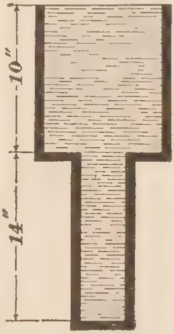


FIG. 4.

EXAMPLE.—A vessel filled with salt water, weighing .037254 pound per cubic inch, has a circular bottom 13 inches in diameter. The top of the vessel is fitted with a piston 3 inches in diameter, on which is laid a weight of 75 pounds; what is the total pressure on the bottom, if the depth of the water is 18 inches?

SOLUTION.—Applying the rule, we have

$13 \times 13 \times .7854 \times 18 \times .037254 = 89.01$ pounds, the pressure due to the weight of the water.

$\frac{75}{3 \times 3 \times .7854} = 10.61$ pounds per square inch, due to the weight on the piston.

$$13 \times 13 \times .7854 \times 10.61 = 1{,}408.29 \text{ pounds.}$$

$$1{,}408.29 + 89.01 = 1{,}497.3 \text{ lb.} = \text{total pressure. Ans.}$$

11. Upward Pressure.—In Fig. 5 is represented a vessel of exactly the same size as that shown in Fig. 4. There is no upward pressure on the surface *c* due to the weight of the water in the large part *c d*, but there is an upward pressure on *c* due to the weight of the water in the small part *b c*. The pressure per square inch due to the weight of the water in *b c* was found to be .5064 pound; the area of the upper surface *c* of the large part *c d* is (6 × 6) − (2 × 2) = 36 − 4 = 32 square inches, and the total upward pressure due to the weight of the water is .5064 × 32 = 16.2 pounds.

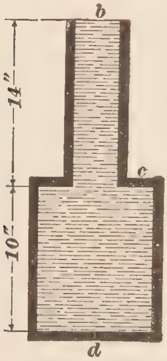


FIG. 5.

If an additional pressure of 10 pounds per square inch were applied to a piston fitting in the top of the vessel, the total upward pressure on the surface *c* would be 16.2 + (32 × 10) = 336.2 pounds.

12. General law for upward pressure:

Rule 2.—*The upward pressure on any submerged horizontal surface equals the weight of a column of the liquid whose base has an area equal to the area of the submerged surface and whose altitude is the distance between the submerged surface and the upper surface of the liquid, increased by the pressure per unit of area on the upper surface of the fluid multiplied by the area of the submerged surface, in case of any pressure on the upper surface.*

EXAMPLE.—A horizontal surface 6 inches by 4 inches is submerged in a vessel of water 26 inches below the upper surface; if the pressure on the water is 16 pounds per square inch, what is the total upward pressure on the horizontal surface?

SOLUTION.—Applying the rule, we get 6 × 4 × 26 × .03617 = 22.57 pounds for the upward pressure due to the weight of the water, and 6 × 4 × 16 = 384 pounds for the upward pressure due to the outside pressure of 16 pounds per square inch.

Therefore, 384 + 22.57 = 406.57 lb. = the total upward pressure. Ans.

13. Lateral (Sidewise) Pressure.—Suppose the top of the vessel shown in Fig. 6 is 10 inches square and that the projections at *a* and *b* are 1 inch square.

The pressure per square inch on the bottom of the vessel due to the weight of the liquid is 1 × 1 × 18 × the weight of a cubic inch of the liquid.

The pressure at a depth equal to the distance of the upper surface of *b* is 1 × 1 × 17 × the weight of a cubic inch of the liquid.

Since both of these pressures are transmitted in every direction, they are also transmitted laterally (sidewise), and the *pressure per unit of area on the projection b is a mean between the two*, and equals $1 \times 1 \times \frac{17 + 18}{2}$ × the weight of a cubic inch of the liquid.

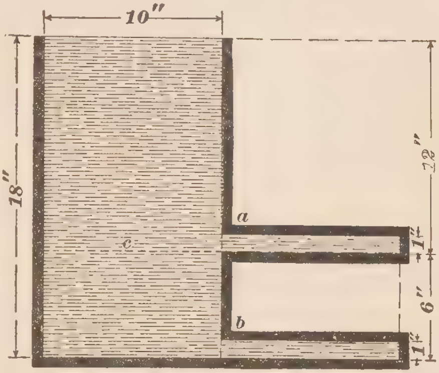


FIG. 6.

To find the lateral pressure on the projection *a*, imagine that the dotted line *c* is the bottom of the vessel; then, the conditions would be the same as in the preceding case, except that the depth is not so great.

The lateral pressure per square inch on *a* is thus seen to be $1 \times 1 \times \frac{11 + 12}{2}$ × the weight of a cubic inch of the liquid.

14. General law for lateral pressure:

Rule 3.—*The pressure on any vertical surface due to the weight of the liquid is equal to the weight of a column of the liquid whose base has the same area as the vertical surface and whose altitude is the depth of the center of gravity of the vertical surface below the upper surface of the liquid.*

Any additional pressure is to be added, as in the previous cases.

EXAMPLE.—A well 3 feet in diameter and 20 feet deep is filled with water; what is the pressure on a strip of wall 1 inch wide, the top of which is 1 foot from the bottom of the well? What is the pressure on the bottom? What is the upward pressure per square inch 2 feet 6 inches from the bottom?

SOLUTION.—Applying the rule, the area of the strip is equal to its length (= circumference of well) multiplied by its height. The length = 36 × 3.1416 = 113.1 inches; height = 1 inch; hence, area of strip = 113.1 × 1 = 113.1 square inches. Depth of center of gravity of strip = (20 − 1) feet + ½ inch, the half width of strip = 228½ inches. Consequently, 113.1 × 228.5 × .03617 = 934.75 lb. = the total pressure on the strip. Ans.

The pressure on each square inch of the strip is $\frac{934.75}{113.1} = 8.265$ lb., nearly. Ans.

36 × 36 × .7854 × 20 × 12 × .03617 = 8,836 lb., the pressure on the bottom. Ans.

20 − 2.5 = 17.5. 1 × 17.5 × 12 × .03617 = 7.596 lb., the upward pressure per square inch 2 ft. 6 in. from the bottom. Ans.

15. The effects of lateral pressure are illustrated in Fig. 7. In the figure, *e* is a tall vessel having a stop-cock near its base and arranged to float upon the water, as shown. When this vessel is filled with water, the lateral pressures at any two points of the surface of the vessel opposite each other are equal. Being equal and acting in opposite directions, they destroy each other, and no motion can result; but, if the stop-cock is opened, there will be no resistance to the pressure acting on that part, and the water will flow out; at the same time, the reaction of the jet issuing from the stop-cock causes the vessel to move backwards through the water in a direction opposite to that of the issuing jet.

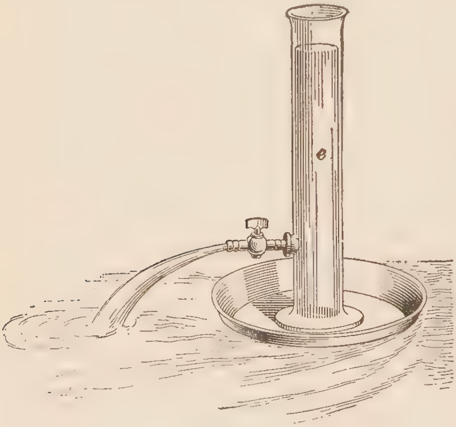


FIG. 7.

16. Liquids Influenced by Gravity.—Since the pressure on the bottom of a vessel due to the weight of the liquid is dependent only on the height of the liquid and not on the shape of the vessel, it follows that if a vessel has a number of radiating tubes (see Fig. 8) the water in each tube will be on the same level, no matter what may be the shape of the tubes. For, if the water were higher in one tube than in the others, the downward pressure on the bottom due to the height of the water in this tube would be greater than that due to the height of the water in the other tubes. Consequently, the upward pressure would also be greater; the equilibrium would be destroyed and the water would flow from this tube into the vessel and rise in the other tubes until it was at the same level in all, when it would be in equilibrium. This principle is expressed in the familiar saying, *water seeks its level.*

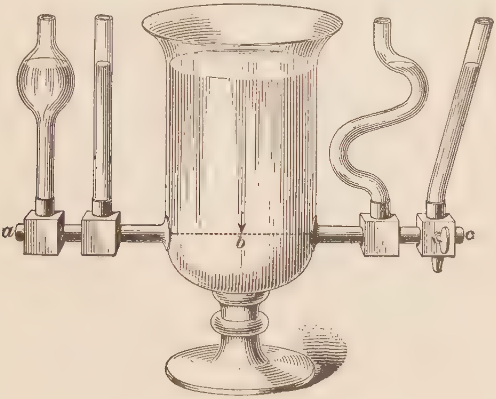


FIG. 8.

An application of this principle is the glass water gauge used for showing the level of the water in a steam boiler or tank.

17. The above principle explains why city water reservoirs are located on high elevations and why water on leaving the hose nozzle spouts so high.

If there were no resistance by friction and air, the water would spout to a height equal to the level of the water in the reservoirs. If a long, vertical pipe, whose length was equal to the vertical distance between the nozzle and the level of the water in the reservoir, were attached to the nozzle, the water would just reach the end of the pipe. If the pipe were lowered slightly, the water would trickle out.

Fountains, canal locks, and artesian wells are examples of the application of this principle.

EXAMPLE.—The water level in a city reservoir is 150 feet above the level of the street; what is the pressure of the water per square inch on the hydrant?

SOLUTION.—Applying rule **1**, $1 \times 150 \times 12 \times .03617 = 65.106$ lb. per sq. in. Ans.

NOTE.—In measuring the height of the water to find the pressure that it produces, the **vertical** height, or distance, between the level of the water and the point considered is always taken. This vertical height is called the **head.**

The weight of a column of water 1 inch square and 1 foot high is $62.5 \div 144 = .434$ pound, nearly. Hence, if the depth (head) be given, the pressure per square inch may be found by multiplying the depth in feet by .434. The constant .434 is the one ordinarily used in practical calculations.

18. In Fig. 9, let the area of the piston a be 1 square inch; of b, 40 square inches. According to Pascal's law, 1 pound placed on a will balance 40 pounds placed on b.

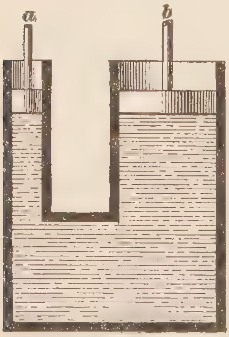


FIG. 9.

Suppose that a moves downwards 10 inches, then 10 cubic inches of water will be forced into the tube b. This will be distributed over the entire area of the tube b, in the form of a cylinder whose cubical contents must be 10 cubic inches, whose base has an area of 40 square inches, and whose altitude must be $\frac{10}{40} = \frac{1}{4}$ inch; that is, a movement of 10 inches of the piston a will cause a movement of $\frac{1}{4}$ inch in the piston b. This is another illustration of the well-known principle of machines: *The force multiplied by the distance through which it acts equals the weight multiplied by the distance through which it moves*, since, if 1 pound on the piston a represents the force P, the equivalent weight W on b may be obtained from the equation $W \times \frac{1}{4} = P \times 10$, whence $W = 40\,P = 40$ pounds.

Another familiar fact is also recognized, for the velocity ratio of P to W is $10 : \frac{1}{4}$, or 40; and since in any machine the weight equals the force multiplied by the velocity ratio, $W = P \times 40$, and when $P = 1$, $W = 40$. An interesting application of this principle is the hydraulic jack, by the aid of which one man can lift a very large load.

19. **A Watson-Stillman hydraulic jack** is shown in section in Fig. 10 (*a*). In this illustration, *a* is a lever that is depressed by the operator when he desires to raise the load. This lever freely fits a rectangular hole or socket in a shaft *b* that extends clear through the ram head. The shaft *b* carries the crank *c* to which the piston *d* is hinged. A small valve is placed in the center of the piston and serves to open or close communication between the water space above and below the piston. The lower end of the ram is fitted with a small check valve, which normally is held closed by the small spring shown beneath the valve. The inside of the jack is filled with a mixture of alcohol and water, about 2 parts of alcohol to 3 parts of water, which prevents the freezing of the liquid when the jack is used in cold weather. The operation is as follows:

The jack being clear down, as shown in Fig. 10 (*a*), it is placed under the load and the operator alternately depresses and raises the lever through its full range, the motion being limited by a projection on the under side of the lever. The slightest depression of the lever closes the valve in the piston. The downward movement of the piston continuing, the water between the bottom of the piston and the end of the ram is subjected to a pressure that opens the valve in the end of the ram. This allows the water under pressure to pass into the space beneath the ram. The pressure so transmitted to the lower end of the ram forces it upwards. The lever having been depressed to its limit is again raised. This raises the piston, causes the piston valve to open, and allows the water above the piston to flow beneath it. The pressure of the water below the ram promptly closes the ram valve as soon as the piston commences to move upwards. The operation may now be repeated as often as required.

As will readily be seen, the ram cannot descend under the weight of the load, as the only water passage leading from

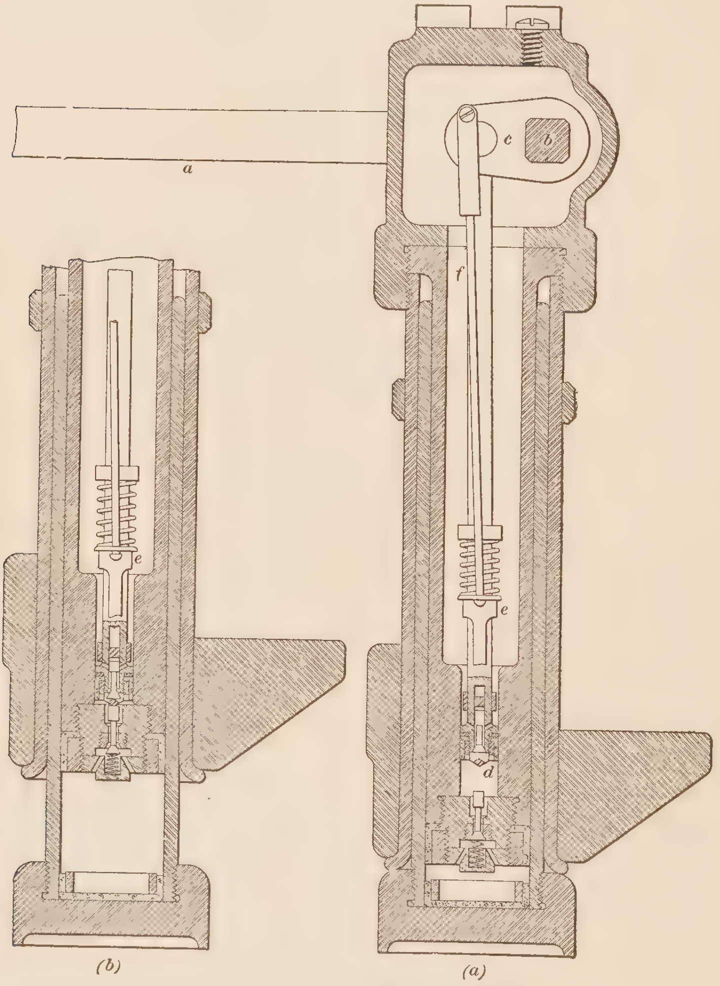


Fig. 10.

the space below the ram is closed by the ram valve, which is kept tightly closed by the pressure of the water beneath it and also by the spring. Hence, in order to lower the ram, this ram valve must be opened by some means. As previously mentioned, the lever has a projection on its under side to limit its travel and the travel of the piston. This projection is made of such a length that when the lever is down as far as it will go, the piston is still a short distance from the upper end of the ram valve stem.

In order to lower the ram, the lever is withdrawn from its socket and inserted again with the projection upwards. It can now be depressed far enough to cause the end of the piston to strike and open the ram-head valve, as shown in Fig. 10 (*b*), where the parts of the jack are shown in the relative positions occupied while lowering. But before the water in the bottom of the ram can escape to the top, the valve in the piston must also be opened. This is accomplished by a heavy spring that forces the sleeve *e* downwards. The lower end of the sleeve has a cotter working in a slot in the piston rod; the cotter bearing against the top of the valve forces the latter downwards. The sleeve is connected to the crank *c* by the so-called lowering wire *f*, in such a manner that the cotter will not strike the piston valve while the jack is raising the load. The ram will descend as long as the lever is kept hard down; it can be stopped instantly by raising the lever slightly.

20. If the area of the piston is 1 square inch and the area of the ram is 10 square inches, the velocity ratio will be 10. If the length of the lever between the hand of the operator and the fulcrum is 10 times the length between the fulcrum and the piston, the velocity ratio of the lever will be 10, and the total velocity ratio of the hand to the piston will be $10 \times 10 = 100$. Taking the weight of the operator at 150 pounds, his whole weight to be thrown on the lever, the weight that can be raised is $150 \times 100 = 15{,}000$ pounds, or $7\frac{1}{2}$ tons. But if the average movement of the hand is 4 inches per stroke, it will require $\frac{100}{4} = 25$ strokes to raise the load on the jack a distance of 1 inch, and it is again seen that what is gained in force is lost in speed.

21. Other applications of this principle are seen in various hydraulic machines used in boiler shops. A familiar example is the hand test pump used in testing boilers under hydraulic pressure.

Example 1.—A vertical cylinder is tested for the tightness of its heads by filling it with water. A pipe whose inside diameter is ¼ inch and whose length is 20 feet is screwed into a hole in the upper head and is then filled with water. What is the pressure per square inch on each head if the cylinder is 40 inches in diameter and 60 inches long?

Solution.— $40^2 \times .7854 = 1{,}256.64$ square inches = area of head.

$1 \times 60 \times .03617 = 2.17$ pounds pressure per square inch on the bottom head due to the weight of the water in the cylinder.

$(\frac{1}{4})^2 \times .7854 = .04909$ square inch, the area of the pipe.

$.04909 \times 20 \times 12 \times .03617 = .426$ pound = the weight of water in the pipe = the pressure on a surface area of .04909 square inch.

The pressure per square inch due to the water in the pipe is $\frac{1}{.04909} \times .426 = 8.68$ lb. per sq. in. upon the upper head. Ans.

The total pressure per square inch on the lower head is $8.68 + 2.17 = 10.85$ lb. Ans.

Example 2.—In the other example, if the pipe be fitted with a piston weighing ¼ pound and a 5-pound weight be laid on it what is the pressure per square inch on the upper head?

Solution.—In addition to the pressure of .426 pound on the area of .04909 square inch, there is now an additional pressure on this area of $5 + \frac{1}{4} = 5.25$ pounds, and the total pressure on this area is $.426 + 5.25 = 5.676$ pounds. $\frac{1}{.04909} \times 5.676 = 115.6$ lb. = the pressure per square inch. Ans.

22. When calculating the weight that can be raised with a hydraulic jack, no allowance was made for the power lost in overcoming the friction between the cup leathers of the piston and the ram, and the cup leather of the ram and the cylinder; this varies according to the condition of the leathers, and, of course,. the smoothness of the ram and cylinder; when the leathers are in good condition, the loss is about 5 per cent. of the total pressure on the ram; when the leathers are old, stiff, and dirty, the loss may amount to 15 per cent. or more.

SPECIFIC GRAVITY.

23. The **specific gravity of a body** is the ratio between its weight and the weight of a like volume of water.

24. Since gases are so much lighter than water, it is usual to take the specific gravity of a gas as the ratio between the weight of a certain volume of the gas and the weight of the same volume of air.

EXAMPLE.—A cubic foot of cast iron weighs 450 pounds; what is its specific gravity, a cubic foot of water weighing 62.42 pounds?

SOLUTION.—According to the definition,

$$\frac{450}{62.42} = 7.21. \quad \text{Ans.}$$

TABLE I.

SPECIFIC GRAVITY AND WEIGHT PER CUBIC FOOT OF VARIOUS METALS.

Substance.	Specific Gravity.	Weight per Cubic Foot. Pounds.
Platinum	21.50	1,343.8
Gold	19.50	1,218.8
Mercury	13.60	850.0
Lead (cast)	11.35	709.4
Silver	10.50	656.3
Copper (cast)	8.79	549.4
Brass	8.38	523.8
Wrought Iron	7.68	480.0
Cast Iron	7.21	450.0
Steel	7.84	490.0
Tin (cast)	7.29	455.6
Zinc (cast)	6.86	428 8
Antimony	6.71	419.4
Aluminum	2.50	156.3

TABLE II.

SPECIFIC GRAVITY AND WEIGHT PER CUBIC FOOT OF VARIOUS WOODS.

Substance.	Specific Gravity.	Weight per Cubic Foot. Pounds.
Ash	.845	52.80
Beech	.852	53.25
Cedar	.561	35.06
Cork	.240	15.00
Ebony (American)	1.331	83.19
Lignum-vitæ	1.333	83.30
Maple	.750	46.88
Oak (old)	1.170	73.10
Spruce	.500	31.25
Pine (yellow)	.660	41.20
Pine (white)	.554	34.60
Walnut	.671	41.90

TABLE III.

SPECIFIC GRAVITY AND WEIGHT PER CUBIC FOOT OF VARIOUS LIQUIDS.

Substance.	Specific Gravity.	Weight per Cubic Foot. Pounds.
Acetic acid	1.062	66.4
Nitric acid	1.217	76.1
Sulphuric acid	1.841	115.1
Muriatic acid	1.200	75.0
Alcohol	.800	50.0
Turpentine	.870	54.4
Sea water (ordinary)	1.026	64.1
Milk	1.032	64.5

25. The specific gravities of different bodies are given in printed tables; hence, if it is desired to know the weight of a body that cannot be conveniently weighed, *calculate its cubical contents and multiply the specific gravity of the body by the weight of a like volume of water, remembering that a cubic foot of water weighs 62.42 pounds.*

EXAMPLE 1.—How much will 3,214 cubic inches of cast iron weigh? Take its specific gravity as 7.21.

SOLUTION.—Since 1 cubic foot of water weighs 62.42 pounds, 3,214 cubic inches weigh

$$\frac{3{,}214}{1{,}728} \times 62.42 = 116.098 \text{ pounds.}$$

Then, $116.098 \times 7.21 = 837.067$ lb. Ans.

TABLE IV.

SPECIFIC GRAVITY AND WEIGHT PER CUBIC FOOT OF VARIOUS GASES AT 32° F. AND UNDER A PRESSURE OF 1 ATMOSPHERE.

Substance.	Specific Gravity.	Weight per Cubic Foot. Pounds.
Atmospheric air	1.0000	.08073
Carbonic acid	1.5290	.12344
Carbonic oxide	.9674	.07810
Chlorine	2.4400	.19700
Oxygen	1.1056	.08925
Nitrogen	.9736	.07860
Smoke (bituminous coal)	.1020	.00815
Smoke (wood)	.0900	.00727
*Steam at 212° F.	.4700	.03790
Hydrogen	.0692	.00559

* The specific gravity of steam at any temperature and pressure compared with air at the same temperature and pressure is .622.

Example 2.—What is the weight of a cubic inch of cast iron?

Solution.— $\frac{62.42}{1,728} \times 7.21 = .26044$ lb. Ans.

One cubic foot of pure distilled water at a temperature of 39.2° Fahrenheit weighs 62.425 pounds, but *the value usually taken in making calculations is 62.5 pounds.*

Example 3.—What is the weight in pounds of 7 cubic feet of oxygen?

Solution.—One cubic foot of air weighs .08073 pound, and the specific gravity of oxygen is 1.1056, compared with air; hence,

$$.08073 \times 1.1056 \times 7 = .62479 \text{ lb., nearly. Ans.}$$

TABLE V.

SPECIFIC GRAVITY AND WEIGHT PER CUBIC FOOT OF VARIOUS SUBSTANCES.

Substance.	Specific Gravity.	Weight per Cubic Foot. Pounds.
Emery	4.00	250
Glass (average)	2.80	175
Chalk	2.78	174
Granite	2.65	166
Marble	2.70	169
Stone (common)	2.52	158
Salt (common)	2.13	133
Soil (common)	1.98	124
Clay	1.93	121
Brick	1.90	118
Plaster of Paris (average)	2.00	125
Sand	1.80	113

BUOYANT EFFECTS OF WATER.

26. In Fig. 11 is shown a 6-inch cube entirely submerged in water. The lateral pressures are equal and in opposite directions. The upward pressure acting on the lower surface of the cube is 6 × 6 × 21 × .03617; the downward pressure acting on the top of the cube is 6 × 6 × 15 × .03617; and the difference is 6 × 6 × 6 × .03617, which equals the volume of the cube in cubic inches times the weight of 1 cubic inch of water. That is, the upward pressure exceeds the downward pressure by the weight of a volume of water equal to the volume of the body.

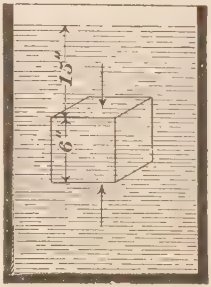


FIG. 11.

This excess of upward pressure over the downward pressure acts against gravity; that is, the water presses the body *upwards* with a greater force than it presses it *downwards;* consequently, *if a body is immersed in a fluid, it will lose in weight an amount equal to the weight of the fluid it displaces.* This is called the **principle of Archimedes,** because it was first stated by him.

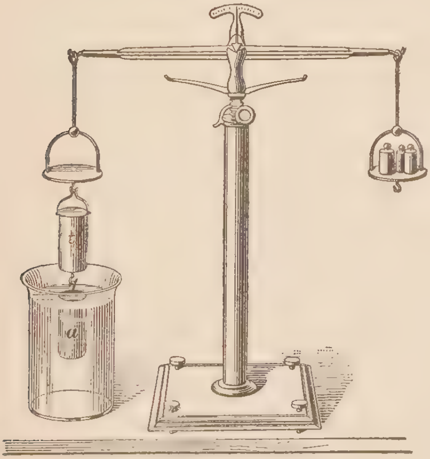


FIG. 12.

27. This principle may be experimentally demonstrated with the beam scales, as shown in Fig. 12. From one scale pan suspend a hollow cylinder of metal *t*, and below that a solid cylinder *a* of the same size as the hollow part of the upper cylinder. Put weights in the other scale pan until they exactly balance the two cylinders. If *a* be immersed in water, the scale pan containing the weights will descend, showing that *a* has lost some of its weight. Now fill *t* with water, and the volume of water that can be poured into *t* will equal that displaced by *a*. The scale pan that contains the weights will gradually rise until *t* is filled, when the scales again balance.

If a body be lighter than the liquid in which it is immersed, the upward pressure will cause it to rise and project partly out of the liquid, until the weight of the body and the weight of the liquid displaced are equal. If the immersed body be heavier than the liquid, the downward pressure plus the weight of the body will be greater than the upward pressure, and the body will fall downwards until it touches bottom or meets an obstruction. If the weights of equal volumes of the liquid and the body are equal, the body will remain stationary and will be in equilibrium in any position or depth beneath the surface of the liquid.

28. An interesting experiment in confirmation of the above facts may be performed as follows: Drop an egg into a glass jar filled with fresh water. The mean density of the egg being a little greater than that of the water, it will fall to the bottom of the jar. Now, dissolve salt in the water, stirring it so as to mix the fresh and salt water. The salt water will presently become denser than the egg and the egg will rise. Now, if fresh water be poured in until the egg and water have the same density, the egg will remain stationary in any position that it may be placed below the surface of the water.

29. The principle of Archimedes gives a very easy and accurate method of finding the volume of an irregularly shaped body. Thus, subtract its weight in water from its weight in air and divide by .03617; the quotient will be the volume in cubic inches, or divide by 62.5 and the quotient will be the volume in cubic feet.

If the specific gravity of the body is known, its cubical contents can be found by dividing its weight by its specific gravity, and then dividing again by either .03617 or 62.5.

EXAMPLE.—A certain body has a specific gravity of 4.38 and weighs 76 pounds; how many cubic inches are there in the body?

SOLUTION.— $\frac{76}{4.38 \times .03617} = 479.72$ cu. in. Ans.

THE HYDROMETER.

30. Instruments called **hydrometers** are in general use for determining quickly and accurately the specific gravities of liquids and some forms of solids. They are of two kinds, viz.: (1) *Hydrometers of constant weight, as* **Beaume's;** (2) *hydrometers of constant volume, as* **Nicholson's.**

A **hydrometer of constant weight** is shown in Fig. 13. It consists of a glass tube, near the bottom of which are two bulbs. The lower and smaller bulb is loaded with mercury or shot, so as to cause the instrument to remain in a vertical position when placed in the liquid. The upper bulb is filled with air, and its volume is such that the whole instrument is lighter than an equal volume of water.

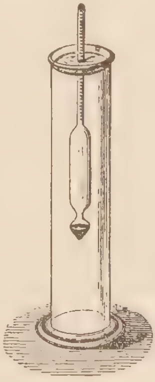


FIG. 13.

The point to which the hydrometer sinks when placed in water is usually marked, the tube being graduated above and below in such a manner that the specific gravity of the liquid can be read directly. It is customary to have two instruments, one with the zero point near the top of the stem for use in liquids heavier than water, and the other with the zero point near the bulb for use in liquids lighter than water.

These instruments are more commonly used for determining the degree of concentration or dilution of certain liquids, as acids, alcohol, milk, solutions of sugar, etc., rather than their actual specific gravities. They are then known as *acidometers*, *alcoholometers*, *lactometers*, *saccharometers*, *salinometers*, etc., according to the use to which they are put.

31. Hydrometers of constant volume are not in common use and are rarely found outside of laboratories.

HYDROKINETICS.

FLOW OF WATER IN PIPES.

32. Experience has demonstrated that for satisfactory work, the flow of water in the suction pipes of boiler feed-pumps and other comparatively small pumps should not exceed 200 feet per minute, and it should not be more than 500 feet in the delivery pipe for a duplex pump, or 400 feet for a single-cylinder pump.

Knowing the volume of water that is to flow through or to be discharged from a pipe in 1 minute, the area of the suction and delivery pipes can readily be determined.

33. The volume of water in cubic feet discharged from a pipe in 1 minute is equal to the velocity in feet per minute times the area of the pipe in square feet. Then, the area of the pipe equals $\frac{\text{volume in cubic feet per minute}}{\text{velocity in feet per minute}}$. As there are 144 square inches in a square foot, the area of the pipe in square inches is $\frac{144 \times \text{volume in cubic feet per minute}}{\text{velocity in feet per minute}}$.

34. If the volume is expressed in gallons per minute, then, as there are 7.48 gallons in a cubic foot, the area of the pipe in square inches will be $\frac{144 \times \text{volume in gallons*}}{7.48 \times \text{velocity in feet per minute}}$. Hence, the following rule,

where n = gallons per minute;
v = velocity in feet per minute;
A = area of pipe in square inches.

Rule 4.—*To find the area of a pipe in square inches, divide 19.25 times the number of gallons per minute by the velocity in feet per minute.*

Or,
$$A = \frac{19.25\,n}{v}.$$

---

* The gallon here referred to is the Winchester or wine gallon of 231 cubic inches capacity and in common use in the United States of America.

EXAMPLE.—If a duplex pump is used, what area of feed-pipe is required for a boiler into which 25 gallons of water per minute is to be pumped?

SOLUTION.—The allowable velocity being 500 feet, by applying rule **4**, we get

$$A = \frac{19.25 \times 25}{500} = .9625 \text{ sq. in.} \quad \text{Ans.}$$

35. The quantity of water, expressed in gallons, that will flow through a given pipe in 1 minute, its velocity being known, is given by the following rule:

Rule 5.—*Multiply the area of the pipe in square inches by the velocity in feet per minute. Divide the product by 19.25.*

Or,
$$n = \frac{A\,v}{19.25}.$$

EXAMPLE.—How many gallons of water per minute will flow through a pipe having an area of 2 square inches, the velocity of flow being 450 feet per minute?

SOLUTION.—Applying the rule just given, we get

$$n = \frac{2 \times 450}{19.25} = 46.75 \text{ gal.} \quad \text{Ans.}$$

36. The velocity with which water will flow through the delivery pipe of a pump when the area of the water cylinder, the area of the delivery pipe, and the piston speed of the pump are known, is given by the following rule,

where v = velocity in feet per minute;
A = area of delivery pipe in square inches;
a = area of water piston in square inches;
S = piston speed in feet per minute.

Rule 6.—*Multiply the area of the water piston by the piston speed, and divide this product by the area of the delivery pipe.*

Or,
$$v = \frac{a\,S}{A}.$$

EXAMPLE.—If the water piston of a pump has an area of 12 square inches and moves at a speed of 100 feet per minute, what will be the velocity of the water in the delivery pipe if the latter has an area of 2 square inches?

SOLUTION.—Applying rule **6**, we get

$$v = \frac{12 \times 100}{2} = 600 \text{ ft. per min.} \quad \text{Ans.}$$

STANDARD PIPE DIMENSIONS.

37. The great majority of the pipes used about steam plants are made of wrought iron and are almost invariably made in accordance with the Briggs standard. It will be noticed that the diameter of the pipe by which it is known to the trade is not the *actual* diameter; hence, in calculating the amount of water that will flow through a pipe, the actual diameter or actual internal area must be taken from Table VI.

38. As wrought-iron pipes are not made in sizes differing from those given in the table, it will be apparent that only in rare instances can a pipe be selected that will have the area calculated by rule **4**. In practice the nearest commercial size of pipe would be selected.

EXAMPLE.—What commercial size of delivery pipe should be used for a single-cylinder pump to deliver 90 gallons of water per minute?

SOLUTION.—For a single-cylinder pump, the velocity of flow should not be more than 400 feet per minute. Then, applying rule **4**, we get

$$A = \frac{19.25 \times 90}{400} = 4.33 \text{ sq. in.}$$

According to Table VI, the commercial size of pipe having an area nearest this is 2½ inches; therefore, a 2½-inch pipe should be used. Ans.

Rule **4** will be found to agree quite closely with the practice of the leading pump manufacturers. In case they should, however, recommend a larger size of pipe, it is advisable to follow their advice.

Pipes made in accordance with the Briggs standard, when below 1 inch nominal size, are butt-welded and proved to 300 pounds per square inch by hydraulic pressure. Pipes above 1 inch are lap-welded and proved to 500 pounds.

TABLE VI.

TABLE OF STANDARD DIMENSIONS OF WROUGHT-IRON WELDED PIPES.

Nominal Diameter. Inches.	Actual Internal Diameter. Inches.	Actual Internal Area. Square Inches.	Actual External Diameter. Inches.	Number of Threads Per Inch.
1/8	.27	.057	.40	27
1/4	.36	.104	.54	18
3/8	.49	.192	.67	18
1/2	.62	.305	.84	14
3/4	.82	.533	1.05	14
1	1.05	.863	1.31	11½
1¼	1.38	1.496	1.66	11½
1½	1.61	2.038	1.90	11½
2	2.07	3.355	2.37	11½
2½	2.47	4.783	2.87	8
3	3.07	7.388	3.50	8
3½	3.55	9.887	4.00	8
4	4.07	12.730	4.50	8
4½	4.51	15.939	5.00	8
5	5.04	19 990	5.56	8
6	6.06	28.889	6.62	8
7	7.02	38.737	7.62	8
8	7.98	50.039	8.62	8
9	9.00	63.633	9.62	8
10	10.02	78.838	10.75	8

PIPE FITTINGS.

39. In piping a steam plant, it is rarely possible to run the piping in a straight line, it being usually necessary to introduce one or more elbows or similar fittings to reach the point desired. The effect of **T** and **L** fittings is to increase the resistance to the flow of the water through the pipes, thus requiring the pump to do more work for the same quantity of water delivered.

As pumps for boiler feeding are always built with a large steam cylinder, there is enough excess of power to allow the pump under ordinary conditions to force the required quantity of water through the pipe. On the suction side of the pump, however, the force impelling the water to flow into the pump is quite small, and a very slight resistance will be sufficient to interfere with the flow of the water into the pump. Hence, it is important that the suction pipe be as straight as possible; if it is impossible to make it straight, larger sizes of pipe should be used, or easy bends with as large a radius as possible should be substituted for the elbows. This applies especially to pumps that must lift the water more than 10 feet. It is impossible to lay down any hard-and-fast rules as to what numbers of elbows should not be exceeded in a suction and delivery pipe; judgment will have to be used. Generally speaking, they should be as few as possible.

EXAMPLES FOR PRACTICE.

1. Suppose a cylinder to be filled with water and placed in an upright position. If the diameter of the cylinder is 19 inches and its total length inside 26 inches, what will be the total pressure on the bottom when a pipe $\frac{1}{2}$ inch in diameter and 12 feet long is screwed into the cylinder head and filled with water? The pipe is vertical. Ans. 1,743.2 lb.

2. In the last example, what is the total pressure against the upper head? Ans. 1,476.6 lb.

3. In example 1, a piston is fitted to the upper end of the pipe and an additional force of 10 pounds is applied to the water in the pipe. What is the total pressure (*a*) on the bottom of the cylinder? (*b*) on the upper head? Ans. (*a*) 16,184 lb. (*b*) 15,917 lb.

4. In example 3, what is the pressure per square inch in the pipe 2 inches from the upper cylinder head? Ans. 56.0656 lb. per sq. in.

5. A water tower 80 feet high is filled with water. A pipe 4 inches in diameter is so connected to the side of the tower that its center is 3 feet from the bottom. If the pipe is closed by a flat cover, what is the total pressure against the cover? Ans. 420 lb.

6. In the last example, what is the upward pressure per square inch 10 feet from the bottom of the tower? Ans. 30.3828 lb. per sq. in.

7. A cube of wood, one edge of which measures 3 feet, is sunk until the upper surface is 40 feet below the level of the water; what is the total force that tends to move the cube upwards? Ans. 1,687.5 lb.

8. A body weighs 72 pounds when immersed in water and 321 pounds in air; what is its volume in cubic feet? Ans. 3.984 cu. ft.

9. The specific gravity of a body being 9.5 and its weight 81 pounds, what is its volume in cubic inches? Ans. 235.73 cu. in.

10. What commercial size of pipe should be used for the suction and delivery pipes of a single-cylinder pump to deliver 50 gallons of water per minute? Ans. $\begin{cases} \text{Suction pipe, } 2\frac{1}{2} \text{ in.} \\ \text{Delivery pipe, } 2 \text{ in.} \end{cases}$

11. How many gallons per hour, at a velocity of 400 feet per minute, will flow through a 1-inch pipe? Ans. 1,076 gal., nearly.

12. The water piston of a pump is $3\frac{1}{2}$ inches; the piston speed is 54 feet per minute; and a $1\frac{1}{2}$-inch delivery pipe is used; what is the velocity in the delivery pipe? Ans. 255 ft. per min., nearly.

PNEUMATICS.

PRESSURE OF GASES.

40. Pneumatics is that branch of mechanics that treats of the properties and pressures of gases.

41. The most striking feature of all gases is their great expansibility. *If we inject a quantity of gas, however small, into a vessel, it will expand and fill that vessel.* If a bladder or football be partly filled with air and placed under a glass jar (called a receiver), from which the air has been exhausted, the bladder or football will immediately expand, as shown in Fig. 14. The force that a gas always exerts when confined in a limited

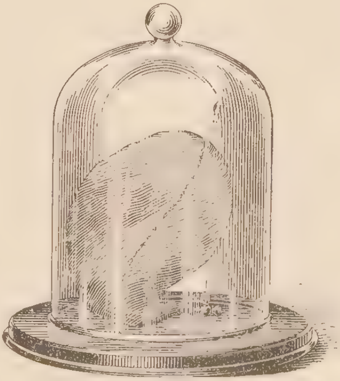


FIG. 14.

space is called *tension*. The word **tension** in this case means pressure, and is only used in this sense in reference to gases.

42. As *water* is the most common type of fluids, so *air* is the most common type of gases. It was supposed by the ancients that air had no weight, and it was not until about the year 1650 that the contrary was proved. A cubic inch of air, under ordinary conditions, weighs .31 grain, nearly. At a temperature of 32° F. and a pressure of 14.7 pounds per square inch, the ratio of the weight of air to water is about 1 : 774; that is, air is only $\frac{1}{774}$ as heavy as water. It has been shown that if a body were immersed in water and weighed less than the volume of water displaced, the body would rise and project partly out of the water. The same is true, to a certain extent, of air. If a vessel made of light material is filled with a gas lighter than air, so that the total weight of the vessel and gas is less than the air they displace, the vessel will rise. It is on this principle that balloons are made.

PRESSURE OF THE ATMOSPHERE.

43. Since air has weight, it is evident that the enormous quantity of air that constitutes the atmosphere must exert a considerable pressure on the earth. This is easily proved by taking a long glass tube closed at one end and filling it with mercury. If the finger be placed over the open end so as to keep the mercury from running out and the tube inverted and placed in a cup of mercury, as shown in Fig. 15, the mercury will fall, then rise, and after a few oscillations will come to rest at a height above the top of the mercury in the cup equal to about 30 inches at sea level.

This height will always be the same under the same atmospheric conditions. Now, if the atmosphere has weight, it must press upon the upper surface of the mercury in the cup with equal intensity upon every square unit, except upon that part of the surface occupied by the tube. In order that there may be equilibrium, the weight of the mercury in the tube must be equal to the pressure of the air upon a portion of the surface of the mercury in the cup equal in area to the inside of the tube. Suppose that the area of the inside of the tube is 1 square inch, then, since mercury is 13.6 times as heavy as water, the weight of the mercurial column is .03617 × 13.6 × 30 = 14.7574 pounds. The actual height of the mercury is a little less than 30 inches, and the actual weight of a cubic inch of distilled water is a little less than .03617 pound. When these considerations are taken into account, the average weight of the mercurial column at the level of the sea, when the temperature is 60° F., is 14.69 pounds, or practically 14.7 pounds. Since this weight, when exerted upon 1 square inch of the liquid in the glass, just produces equilibrium, it is plain that the pressure of the outside air is 14.7 pounds upon every square inch of surface.

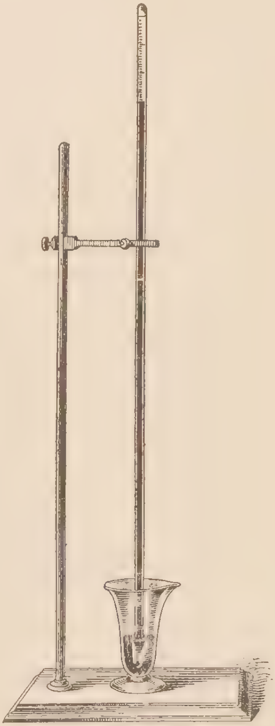


FIG. 15.

44. Vacuum.—The space between the upper end of the tube and the upper surface of the mercury is called a **vacuum,** meaning that it is an entirely empty space and does not contain any substance—solid, liquid, or gaseous. If there were a gas of some kind there, no matter how small the quantity might be, it would expand and fill the space, and its tension would cause the column of mercury to fall and become shorter, according to the amount of gas or air present. The condition then existing in the space would be called a **partial vacuum.**

45. The Measurement of Vacuum.—The degree to which the air has been exhausted from a closed vessel in which there is a partial vacuum is measured by the height to which a mercurial column in a vertical tube, whose top is connected to the vessel, will rise under the pressure of the atmosphere. Thus, when we say that there is a vacuum of 29 inches in a vessel, we mean that enough air has been exhausted from the vessel to enable the surrounding air to support a mercurial column 29 inches high, as shown in Fig. 16, where *A* is the vessel in which there is a partial vacuum. In other words, the pressure in the vessel is less than the pressure of the atmosphere by a column of mercury 29 inches in height.

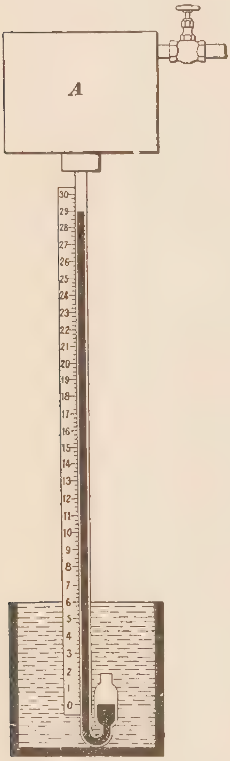


FIG. 16.

When there is a partial vacuum inside a closed vessel, the air remaining in the vessel is under a tension, or pressure, less than its original tension of 14.7 pounds per square inch. This tension, in inches of mercury, is equal to the difference between the height at which the mercurial column connected to the vessel stands and that at which it would stand if the vacuum were perfect. Consider that the mercury column will be in equilibrium when the hydrostatic pressure on its base equals the atmospheric pressure. The hydrostatic pressure on the base is the sum of two pressures: (1) The pressure due to the weight of the mercury column; (2) the pressure in the space above the mercury.

From this it follows that if the atmospheric pressure and the pressure due to the weight of the mercurial column are given, the pressure above the column must be that due to their difference. As the atmosphere will force a mercurial column to a height of 30 inches when there is a perfect vacuum above it, it follows that to find the pressure in a vessel in which there is a partial vacuum, the number of inches of height of the mercurial column is to be subtracted from 30. Thus, if the vacuum in a vessel is 26 inches, the pressure in the vessel is 30 − 26 = 4 inches of mercury.

46. In practice it is convenient to always use the same unit of pressure, which is 1 pound per square inch. We know that 14.7 pounds per square inch will support a mercurial column 30 inches high. Hence, 1 inch of height of a mercurial column represents a pressure of $\frac{14.7}{30} = .49$ pound per square inch. Then, to reduce inches of mercury to pounds per square inch, multiply their number by .49.

EXAMPLE.—A gauge shows a vacuum of 22 inches in a condenser. What is the absolute pressure in the condenser?

SOLUTION.—The pressure, in inches of mercury, is 30 − 22 = 8 inches, or 8 × .49 = 3.92 lb. per sq. in. Ans.

47. The Vacuum Gauge.—For engineering work, the glass tube of Fig. 16 would be a rather inconvenient form of gauge for measuring the vacuum. Hence, special metallic gauges, known as **vacuum gauges,** have been designed. Their dial is graduated to show the degree of vacuum, in inches of mercury. In steam-engineering work, they are used chiefly in connection with condensers for steam engines. It should be borne in mind that a vacuum gauge does not indicate the pressure in the vessel to which it is attached, but instead shows, in inches of mercury, how much the pressure has been *lowered below the atmospheric pressure.* In this respect a vacuum gauge differs essentially from a **pressure gauge,** which shows how much the pressure has been *increased* either above the pressure of a perfect vacuum, which is zero, as in case of a gauge registering the pressure of the atmosphere, or above the atmospheric pressure, as in case of the ordinary pressure gauge.

48. If the tube of Fig. 15 had been filled with a liquid lighter than mercury, the height of the column required to balance the atmospheric pressure would be greater. The height of the column depends on the specific gravity of the liquid. Thus, if the liquid had a specific gravity of 1, as water, its height would be $\frac{30 \times 13.6}{1} = 408$ inches, or 34 feet. This means that if a tube be filled with water, inverted, and placed in a dish of water, in a manner similar to the experiment made with the mercury, the height of the column of water will be 34 feet.

49. The Barometer.—As is well known, the atmosphere does not exert exactly the same pressure at all times; the pressure varies with conditions. As the height of the mercury in the glass tube of Fig. 15 depends chiefly on the atmospheric pressure, it is evident that an instrument constructed on this principle will indicate the varying pressure by the height of the column. Such an instrument is called a **mercurial barometer.**

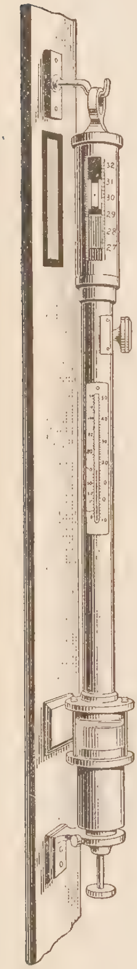


FIG. 17.

50. A standard form of barometer is shown in Fig. 17. The barometer is simply a pressure gauge that registers the pressure of the air. In this case the cup and tube at the bottom are protected by a brass or iron casing. At the top of the tube is a graduated scale. Attached to the casing is an accurate thermometer for determining the temperature of the outside air at the time the barometric observation is taken. This is necessary, since mercury expands when the temperature is increased and contracts when the temperature falls; for this reason a standard temperature is assumed, and all barometer readings are reduced to this temperature. This standard temperature is usually taken at 32° F., at which temperature the height of the mercurial column is 30 inches under normal conditions at sea level. Another correction is made for the altitude of the place above sea level, and a third correction for the effects of capillary attraction.

51. A mercurial barometer is not only a very bulky instrument, but is also quite heavy and, hence, is not very well adapted for transportation. To overcome these drawbacks, a form of portable barometer known as an **aneroid barometer,** has been designed, which operates on a somewhat different principle. Such a barometer is shown in

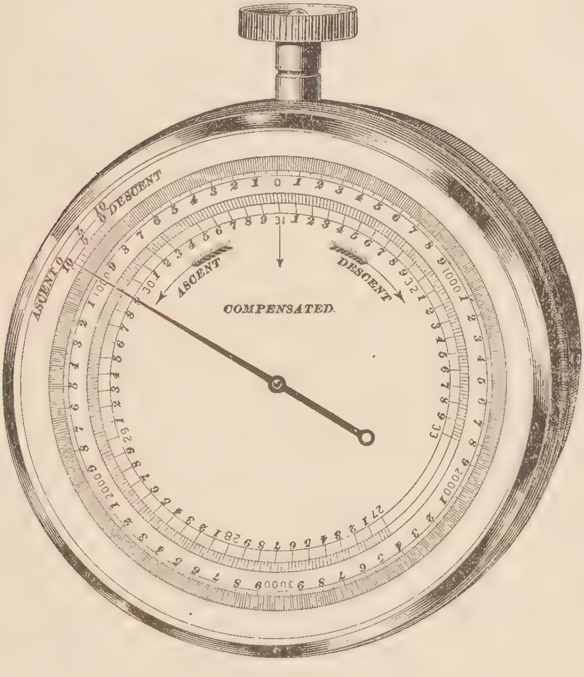


FIG. 18.

Fig. 18. The principal part of the aneroid barometer is a cylindrical, air-tight box of metal closed by a corrugated top of thin, elastic metal. The air is exhausted from the box, which is then sealed. Evidently, the pressure of the air on the outside of the cover will cause the cover to curve inwards, as the space inside of the cover is void of pressure, until the

resistance due to the elasticity of the cover, aided by the resistance of a spring beneath it, is equal to the force exerted by the air. Now, if the air pressure increases, the cover will be curved further inwards; if the air pressure decreases, the elasticity of the cover, aided by the spring beneath, will cause a reduction of its inward curvature. These movements of the cover are transmitted and multiplied by a combination of delicate levers that act on the index hand shown in the figure and cause it to move to the right or left over a graduated scale.

These barometers are compensated (self-correcting) for variations in temperature. The better grade of these instruments have two graduations; the inner graduation corresponds to the graduations of the mercurial barometer; that is, it reads to inches of mercury. The outer graduation represents elevations above and below sea level, the distance between each graduation line representing a difference in elevation of 10 feet.

These instruments are made in various sizes, from the size of a watch up to 8 or 10 inches in diameter. They are very portable, occupying but a small space, are very light, and are quite delicate.

52. Both the mercurial and the aneroid barometers operate on the same general principle; viz., that if two opposite forces act on a body, it will be in equilibrium when the forces are equal and will be set in motion when they are unequal, provided the difference in the magnitude of the two forces is sufficient to overcome the resistance.

The two styles of barometer differ from each other only in the method by which equilibrium is established. In the mercurial barometer, the weight of the mercurial column inside the tube equalizes the outside air pressure; in the aneroid barometer, the outside pressure is equalized by the resistance of the flexible cover and spring beneath it.

53. Variations of Pressure at Different Elevations. With air, as with water, the lower we get the greater is the pressure, and the higher we get the less is the pressure. At the level of the sea, the height of the mercurial column is about 30 inches; at 5,000 feet above the sea, it is 24.7 inches; at 10,000 feet above the sea, it is 20.5 inches; at 15,000 feet, it is 16.9 inches; at 3 miles, it is 16.4 inches; and at 6 miles above the sea level, it is 8.9 inches.

Air being an elastic fluid, the density or weight of the atmosphere also varies with the altitude; that is, a cubic foot of air at an elevation of 5,000 feet above the sea level will not weigh as much as a cubic foot at sea level. This is proved conclusively by the fact that at a height of 3½ miles the mercurial column measures but 15 inches, indicating that half the weight of the entire atmosphere is below that height. It is known that the height of the earth's atmosphere is at least 50 miles; hence, the air just before reaching the limit must be in an exceedingly rarefied state. It is by means of barometers that great heights are measured. The aneroid barometer has the heights marked on the dial, so that they can be read directly. With the mercurial barometer, the heights must be calculated from the reading.

54. Pressure in Different Directions.—The atmospheric pressure is everywhere present and presses all objects in all directions with equal force. If a book is laid upon the table, the air presses upon it in every direction with an equal average force of 14.7 pounds per square inch. It would seem as though it would take considerable force to raise a book from the table, since, if the size of the book were 8 inches by 5 inches, the pressure upon it would be $8 \times 5 \times 14.7 = 588$ pounds; but there is an equal pressure *beneath* the book that counteracts the pressure on the top. It would now seem as though it would require a great force to open the book, since there are two pressures of 588 pounds each acting in opposite directions and tending to crush the book; so it would, but for the fact that there is a layer of air beneath each leaf, which acts upwards and downwards with a pressure of 14.7 pounds per square inch.

If a piece of flat glass be laid upon a flat surface that has been previously moistened with water, it will require considerable force to separate them; this is because the water helps to fill the pores in the flat surface and glass and thus creates a partial vacuum between the glass and the surface, thereby reducing the counter pressure beneath the glass.

55. Tension of Gases.—In Fig. 15, the space above the column of mercury was said to be a vacuum, and it was also stated that if any gas or air were present, it would expand and its tension would force the column of mercury downwards. If sufficient gas be admitted to cause the mercury to stand at 15 inches, the tension of the gas is evidently $\frac{14.7}{2} = 7.35$ pounds per square inch, since the pressure of the outside air, 14.7 pounds per square inch, now balances only 15 instead of 30 inches of mercury; that is, it balances only one-half as much as it would if there were no gas in the tube; therefore, the pressure (tension) of the gas in the tube is 7.35 pounds. If more gas be forced into the tube until the top of the mercurial column is just level with the mercury in the cup, the gas in the tube will then have a tension equal to the outside pressure of the atmosphere. Suppose that the bottom of the tube is fitted with a piston, and that the total length of the inside of the tube is 36 inches. If the piston be shoved upwards so that the space occupied by the gas is 18 inches long instead of 36 inches, the temperature remaining the same as before, it will be found that the tension of the gas within the tube is 29.4 pounds. It will be noticed that the volume occupied by the gas is only half that in the tube before the piston was moved, while the pressure is twice as great, since $14.7 \times 2 = 29.4$ pounds. If the piston be shoved up, so that the space occupied by the gas is only 9 inches instead of 18 inches, the temperature still remaining the same, the pressure will be found to be 58.8 pounds per square inch. The volume has again been reduced one-half and the pressure increased twofold, since $29.4 \times 2 = 58.8$ pounds. The space now occupied by the gas is 9 inches long, whereas, before the piston was moved, it was 36 inches long; as the tube is assumed to be of uniform diameter throughout its length, the volume is now $\frac{9}{36} = \frac{1}{4}$ of its original volume, and its pressure is $\frac{58.8}{14.7} = 4$ times its original pressure. Moreover, if the temperature of the confined gas remains the same, the pressure and volume will always vary in a similar way.

56. Absolute and Gauge Pressures.—From the above explanation, it will be apparent that the pressure in the tube is the pressure above that of a perfect vacuum. Pressures reckoned above vacuum are called **absolute pressures.** The only pressure gauges that indicate absolute pressures are the mercurial and aneroid barometers; ordinary pressure gauges, such as the common steam gauge, indicate pressures above the pressure of the atmosphere. Pressures above that of the atmosphere are commonly called **gauge pressures.** Gauge pressures are changed to absolute pressures by adding 14.7 pounds per square inch to their readings. Truly speaking, the pressure indicated by a barometer, reduced to pounds pressure, should be added to the gauge pressure, since the value 14.7 pounds only represents the mean atmospheric pressure under normal conditions at *sea level.*

57. Mariotte's Law.—The law that states the effect of compressing and expanding gases is called **Mariotte's law** and is as follows:

The temperature remaining the same, the volume of a given quantity of gas varies inversely as the absolute pressure.

The meaning of the law is this: If the volume of a gas be diminished to $\frac{1}{2}$, $\frac{1}{3}$, $\frac{1}{5}$, etc. of its former value, the tension will be increased 2, 3, 5, etc. times, or if the outside pressure be increased 2, 3, 5, etc. times, the volume of the gas will be diminished to $\frac{1}{2}$, $\frac{1}{3}$, $\frac{1}{5}$, etc. of its original volume, the temperature remaining constant. It also means that if a gas is under a certain pressure, and this pressure is diminished to $\frac{1}{2}$, $\frac{1}{4}$, $\frac{1}{10}$, etc. of its original intensity, the volume of the confined gas will be increased 2, 4, 10, etc. times—its tension decreasing at the same rate.

Suppose 3 cubic feet of air to be under a pressure of 60 pounds per square inch in a cylinder fitted with a movable piston; then the product of the volume and pressure is $3 \times 60 = 180$. Let the volume be increased to 6 cubic feet; then the pressure will be 30 pounds per square inch, and $30 \times 6 = 180$, as before. Let the volume be increased to 24 cubic feet; it is then $\frac{24}{3} = 8$ times its original volume, and the pressure is $\frac{1}{8}$ of its original pressure, or $60 \times \frac{1}{8} = 7\frac{1}{2}$ pounds, and $24 \times 7\frac{1}{2} = 180$, as in the two preceding cases. It will now be noticed that if a gas be enclosed within a confined space and allowed to expand without losing any heat, *the product of the pressure and the corresponding volume for any one position of the piston is the same as for any other position.* If the piston were forced inwards so as to compress the air, the same results would be obtained.

58. Application of Mariotte's Law.—If the volume of the vessel and the pressure of the gas are known, and it is desired to know the pressure after the first volume has been changed:

Rule 7.—*Divide the product of the first, or original, volume and pressure by the new volume; the result will be the new pressure.*

Or, let p = original absolute pressure;
p_1 = final absolute pressure;
v = volume corresponding to the pressure p;
v_1 = volume corresponding to the pressure p_1.

Then,
$$p_1 = \frac{p\,v}{v_1}.$$

Example.—At the point of cut-off in a steam engine, the amount of steam in the cylinder is 862 cubic inches. The pressure at this point is 120 pounds per square inch. What will be the pressure of the steam when the piston has reached the end of its stroke and the volume is 1,800 cubic inches?

SOLUTION.—Applying the rule, we get

$$p_1 = \frac{862 \times 120}{1,800} = 57.47 \text{ lb. per sq. in. absolute.} \quad \text{Ans.}$$

59. If it is required to determine the volume after a change in the pressure:

Rule 8.—*Divide the product of the original volume and pressure by the new pressure; the result will be the new volume.*

Or, using the same letters as before,

$$v_1 = \frac{p\,v}{p_1}.$$

EXAMPLE 1.—At the commencement of compression, the volume of the steam is 380 cubic inches and the pressure is 18 pounds per square inch. At the end of compression, the pressure is 112 pounds per square inch. What is the final volume?

SOLUTION.—Applying the rule, we get

$$v_1 = \frac{380 \times 18}{112} = 61.07 \text{ cu. in.} \quad \text{Ans.}$$

EXAMPLE 2.—A vessel contains 10 cubic feet of air at a pressure of 15 pounds per square inch and has 25 cubic feet of air of the same pressure forced into it; what is the resulting pressure?

SOLUTION.—The original volume $= 10 + 25 = 35$ cubic feet; the original pressure is 15 pounds per square inch; the final volume is 10 cubic feet. Hence, applying rule **7**, we get

$$p_1 = \frac{35 \times 15}{10} = 52.5 \text{ lb. per sq. in. absolute.} \quad \text{Ans.}$$

It must be remembered that in the preceding examples the temperature is supposed to remain constant. When the temperature changes during expansion and compression, the problem becomes a rather difficult one and cannot be solved by ordinary arithmetic.

EXAMPLES FOR PRACTICE.

1. A vessel contains 25 cubic feet of gas at a pressure of 18 pounds per square inch; if 125 cubic feet of gas having the same pressure are forced into the vessel, what will be the resulting pressure?

Ans. 108 lb. per sq. in.

2. The volume of steam in the cylinder of a steam engine at cut-off is 1.35 cubic feet and the pressure is 85 pounds per square inch; the pressure at the end of the stroke is 25 pounds per square inch. What is the new volume? Ans. 4.59 cu. ft.

3. A receiver contains 180 cubic feet of gas at a pressure of 20 pounds per square inch; if a vessel holding 12 cubic feet be filled from the larger vessel until its pressure is 20 pounds per square inch, what will be the pressure in the larger vessel?

Ans. $18\frac{2}{3}$ lb. per sq. in.

4. A spherical shell has a part of the air within it removed, forming a partial vacuum; if the outside diameter of the shell is 18 inches and the pressure of the air within is 5 pounds per square inch, what is the total pressure tending to crush the shell? Ans. 9,873.42 lb.

PNEUMATIC MACHINES.

THE AIR PUMP.

60. *The* **air pump** *is an instrument for removing air from a given space.* A section of the principal parts is shown in Fig. 19 and the complete instrument in Fig. 20. The closed vessel *R* is called the **receiver,** and the space

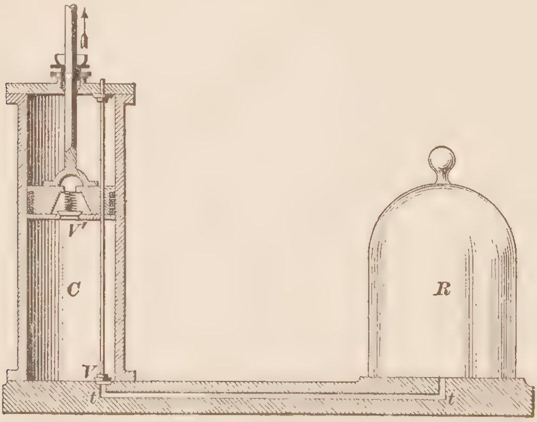


FIG. 19.

that it encloses is that from which it is desired to remove the air. It is usually made of glass, and the edges are ground so as to be perfectly air-tight. When made in the form shown, it is called a **bell-jar receiver.** The receiver rests upon a horizontal plate, in the center of which is an opening communicating with the pump cylinder *C* by means of the passage *t t*. The pump piston accurately fits the cylinder, and has a valve *V′* opening upwards. Where the passage *t t* joins the cylinder, another valve *V* is placed, which also opens upwards. When the piston is raised, the valve *V′* closes, and, since no air can get into the cylinder from above, the piston leaves a vacuum behind it. The pressure upon *V* being now removed, the tension of the air in the receiver *R* causes *V* to rise; the air in the receiver and passage *t t* then expands so as to occupy the additional space provided by the upward movement of the piston.

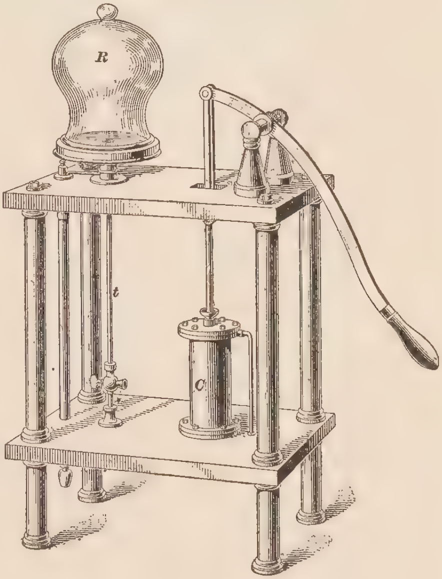


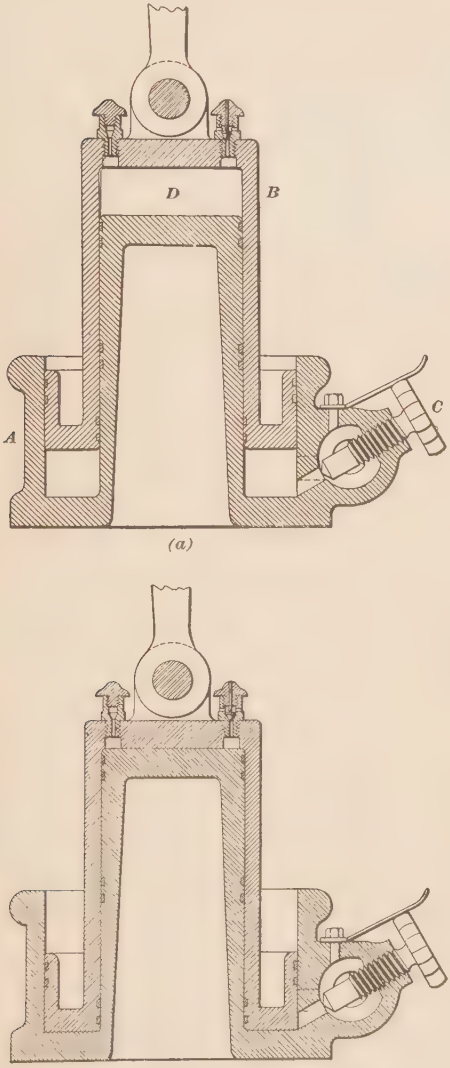
FIG. 20.

The piston is now pushed down, the valve *V* closes, the valve *V′* opens, and the air in *C* escapes. The lower valve *V* is sometimes supported, as shown in Fig. 19, by a metal rod passing through the piston and fitting it somewhat tightly. When the piston is raised or lowered, this rod moves with it. A button near the upper end of the rod confines its motion within very narrow limits, the piston sliding upon the rod during the greater part of the journey. In the complete form of the instrument shown in Fig. 20, communication between receiver and pump is made by means of the tube t.

61. Degrees and Limits of Exhaustion.—Suppose that the volume of R and t together is four times that of C, Fig. 19, and that there are, say, 200 grains of air in R and t, and 50 grains in C when the piston is at the top of the cylinder. At the end of the first stroke, when the piston is again at the top, 50 grains of air in the cylinder C will have been removed, and the 200 grains in R and t will occupy the space R, t, and C. The ratio between the sum of the spaces R and t and the total space $R + t + C$ is $\frac{4}{5}$; hence, $200 \times \frac{4}{5} = 160$ grains = the weight of air in R and t after the first stroke. After the second stroke, the weight of the air in R and t would be $(200 \times \frac{4}{5}) \times \frac{4}{5} = 200 \times (\frac{4}{5})^2 = 200 \times \frac{16}{25} = 128$ grains. At the end of the third stroke, the weight would be $[200 \times (\frac{4}{5})^2] \times \frac{4}{5} = 200 \times (\frac{4}{5})^3 = 200 \times \frac{64}{125} = 102.4$ grains. At the end of n strokes the weight would be $200 \times (\frac{4}{5})^n$. It is evident that *it is impossible to remove all the air that is contained in R and t by this method.* It requires an exceedingly good air pump to reduce the tension of the air in R to $\frac{1}{50}$ inch of mercury. When the air has become so rarefied as this, the valve V' will not lift, and, consequently, no more air can be exhausted.

62. The Dashpot.—The pressure of the atmosphere is utilized in Corliss engines to close the steam valves rapidly. For this purpose a so-called **dashpot** is used. The dashpot of a Bullock-Corliss engine is shown in two positions in Fig. 21. It consists of the base A, which is fastened to the floor, and a plunger B pivoted to the end of a crank-arm keyed to the stem of the steam valve. The base is accurately turned and bored to fit the plunger; packing rings are fitted to both the plunger and the base in order to make an air-tight joint.

The annular space around the central stationary piston is in communication with the outside air by a small passage, the opening of which can be increased or decreased, at will, by means of the valve *C*. In operation, the plunger is first in the position shown in Fig. 21 (*b*). It is then picked up by the valve gear of the engine and drawn up until it is in the position shown in Fig. 21 (*a*). The large piston on the lower end of the plunger is in communication with the atmosphere through the valve *C*, and hence the pressure of the atmosphere on both sides of the piston is equal. The upper part of the plunger, however, has an equal pressure on both its sides only when it is down, as in Fig. 21 (*b*). When it is drawn up, the air in the space *D* expands and a partial vacuum is formed. The valve gear next releases the plunger, and as the atmospheric pressure is much greater than the pressure within *D*, the plunger rapidly descends. During its descent, the large piston at the end of the plunger gradually compresses the air in the annular space beneath it, and is



(*a*)

(*b*)

FIG. 21.

thus gradually brought to rest. The valve *C* serves to regulate the amount of compression and at the same time admits air during the up stroke of the plunger.

THE SIPHON.

63. Theory of the Siphon.—The action of the **siphon** illustrates the effect of atmospheric pressure. It is simply a bent tube having unequal branches, open at both ends, and is used to convey a liquid from a higher point to a lower over an intermediate point that is higher than either. In Fig. 22, *A* and *B* are two vessels, *B* being lower than *A*, and *A C B* is the bent tube, or siphon. Suppose this tube to be filled with water and placed in the vessels as shown, with the short branch *A C* in the vessel *A*. The water will flow from the vessel *A* into the vessel *B* as long as the level of the water in *B* is below the level of the water in *A* and the level of the water in *A* is above the lower end of the tube *A C*. The atmospheric pressure on the surfaces of *A* and *B* tends to force the water up the tubes *A C* and *B C*. When the siphon is filled with water, each of these pressures is counteracted in part by the pressure of the water in that branch of the siphon that is immersed in the water on which the pressure is exerted. The atmospheric pressure opposed to the weight of the longer column of water will, therefore, be resisted more strongly than that opposed to the weight of the shorter column; consequently, the pressure exerted on the shorter column will be greater than that on the longer column, and this excess of pressure will produce motion.

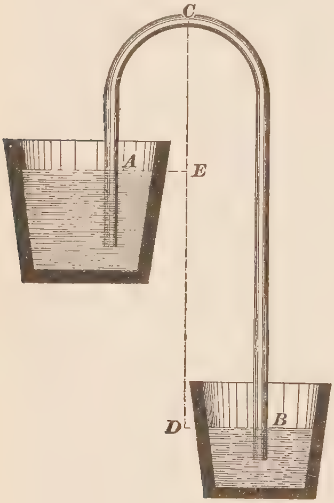


FIG. 22.

64. Let A = the area of the tube;

$h = D\,C$ = the vertical distance between the surface of the water in B and the highest point of the center line of the tube;

$h_1 = E\,C$ = the distance between the surface of the water in A and the highest point of the center line of the tube.

The weight of the water in the short column is $.03617 \times A\ h_1$, and the resultant atmospheric pressure tending to force the water up the short column is $14.7 \times A - .03617 \times A\ h_1$. The weight of the water in the long column is $.03617\ A\ h$, and the resultant atmospheric pressure tending to force the water up the long column is $14.7\ A - .03617\ A\ h$. The difference between these two is $(14.7\ A - .03617\ A\ h_1) - (14.7\ A - .03617\ A\ h) = .03617\ A\ (h - h_1)$. But $h - h_1 = E\ D$ = the difference between the levels of the water in the two vessels. In the above, h and h_1 were taken in inches and A in square inches.

It will be noticed that the short column must not be higher than 34 feet for water, or the siphon will not work, since the pressure of the atmosphere will not support a column of water that is higher than 34 feet.

65. Fig. 23 shows the actual construction of a siphon. It is desired to convey the water from D to E. The point of discharge must always be lower than the point from which the water is taken, otherwise a siphon cannot work. The siphon consists of ordinary iron pipe jointed in any convenient manner so as to be air-tight. It has three valves A, B, and C. The suction end is provided with a perforated strainer c to keep out large particles of foreign matter.

In order to start the siphon, it is necessary to remove the air in the pipe. This is done by closing the valves A and B and opening the valve C, which should always be located at the highest point of the siphon. Water is then poured into the funnel above the valve C until the pipe is filled. When no more water can be poured in, that is, when all the air has been driven out, the valve *C* is closed and the valves *A* and *B* are opened; the siphon will now start.

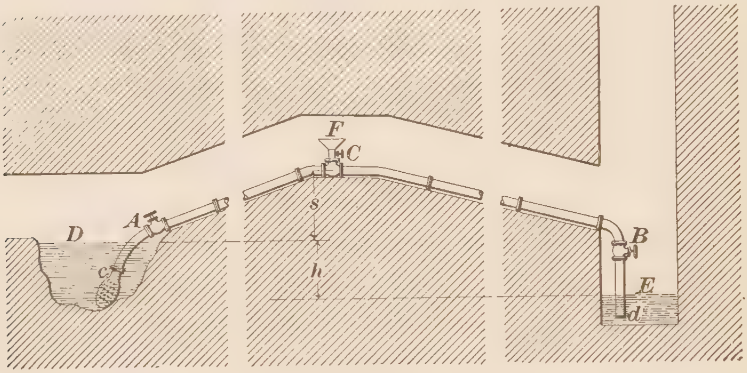


FIG. 23.

In practice, it has been found that the distance *s* must not exceed 28 feet at sea level, and that the siphon will work more satisfactory if it is less. The greater the distance *h* between the two water levels, the better the siphon will work.

66. In order that a siphon will work properly, it is necessary that air should be kept out of the pipe, or, if it does get in, means should be provided for its escape. The joints of the pipe must be perfectly air-tight. But some air will always be carried in with the water, and this air will collect at the highest point. The bad effects of this can be minimized by having a straight horizontal pipe at the highest point instead of a sharp bend.

67. A device that will remove the air is shown in Fig. 24. Here *A* is an air-tight vessel connected with the siphon by two small pipes *B* and *C*. The pipe *B* extends nearly to the top of *A*, while the pipe *C* barely enters the bottom. Each pipe has a valve *D* and *E*. A valve *F* and funnel *G* are placed on top of the vessel. When the siphon ceases to flow, which is an indication that air has collected (it is here assumed that the suction end has not become uncovered), the valves *D* and *E* are closed and the valve *F* is opened. The vessel is now filled with water, the valve *F* is closed, and *D* and *E* are opened. The water will now flow through *C* into the siphon and force out the air collected there, which passes through *B* to the top of *A*. When all air is out of the siphon, *D* and *E* are shut and *F* is opened. The vessel *A* is now filled with water, *F* is shut, *D* and *E* are opened and left open. Any air that enters the siphon will, instead of collecting at *H*, ascend the pipe *B* and force out a small amount of water through *C*. This will continue until *A* is filled with air, when the valves *D* and *E* should be closed and *A* refilled. This arrangement may also be used to fill the siphon for the purpose of setting it to work.

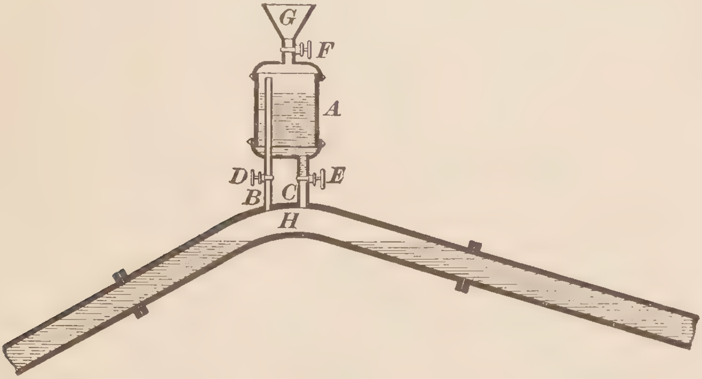


FIG. 24.

The highest point of the water in *A* should not be more than 28 feet at sea level above the water level at the suction end.

PUMPS.

68. A **pump** is a machine for conveying liquids from one level to a higher level or for performing work equivalent to such an operation.

69. Classification of Pumps.—Pumps may be divided into three general divisions, according to the service they perform, viz., *suction pumps*, *lifting pumps*, and *force pumps.* They may also be divided into two general classes, *single-acting* and *double-acting pumps*, according as they take water on one side or on both sides of the water piston. According to the arrangement of the pump cylinders, they are classified as *simple*, *duplex*, or *triplex pumps.* As pumps displace the liquids in various ways, they may also be divided according to the method of displacement, into *reciprocating*, *centrifugal*, and *rotary pumps.* Reciprocating pumps only will be considered here.

70. The Suction Pump.—A section of an ordinary suction pump is shown in diagrammatic form in Fig. 25. Suppose the piston, or **bucket** as it is commonly termed, to be at the bottom of the cylinder and to be just on the point of moving upwards in the direction of the arrow. As the piston rises it leaves a partial vacuum behind it, and the atmospheric pressure on the surface of the water in the well causes it to rise in the pipe *P* for the same reason that the mercury rises in the barometer tube. The water rushes up the pipe and lifts the suction valve *V*, filling the empty space in the cylinder *B* caused by the displacement of the piston. When the piston has reached the end of its stroke, the water entirely fills the space between the bottom of the piston and the

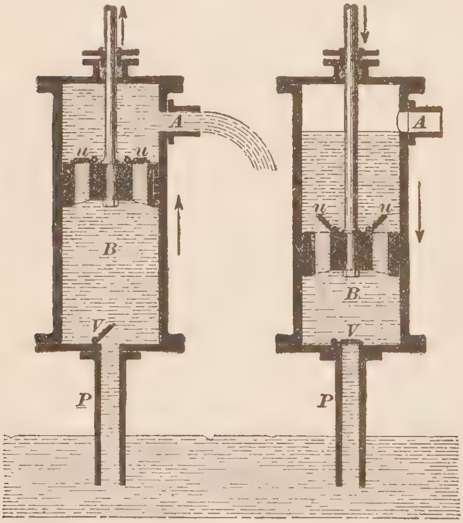


FIG. 25.

bottom of the cylinder, and also the pipe *P*. The instant the piston begins its down stroke, the water in the chamber *B* begins to flow back into the well, and its downward flow forces the valve *V* to its seat, thus preventing any further escape of the water. As the piston descends, the water must give way to it, and since the suction valve *V* is closed, the bucket valves *u*, *u* must open, and thus allow the water to pass through the piston, as shown in the right-hand figure. When the piston has reached the end of its downward stroke and commenced its upward movement again, the water flowing through the piston quickly closes the valves *u*, *u*. All the water resting on the top of the piston is then lifted by the piston on its upward stroke and discharged through the spout *A*; the valve *V* again opens and the water fills the space below the piston, as before.

71. It is evident that the distance between the piston when at the top of its stroke and the surface of the water in the well must not exceed 34 feet, the highest column of water that the pressure of the atmosphere will sustain, since, otherwise, the water in the pipe would not rise and fill the cylinder as the piston ascended. In practice, this distance should not exceed 28 feet. This is due to the fact that there is a little air left between the bottom of the piston and the bottom of the cylinder, a little air leaks through the valves, which are not perfectly air-tight, and a pressure is needed to raise the valve against its own weight, which, of course, acts downwards.

There are many varieties of the suction pump, differing principally in the construction of the valves and piston, but the principle is the same in all.

72. The Lifting Pump.—In some cases it is desired to raise water higher than it can be forced by the pressure of the atmosphere into the chamber of a simple suction pump, such as is shown in Fig. 25. To accomplish this the pump chamber with its bucket and valves are set at a distance above the supply not exceeding that to which the air will successfully force the water. A closed pipe *P*′, Fig. 26, called the *delivery*, or *discharge*, *pipe*, is then led from the upper part of the chamber to the point where it is desired to deliver the water. Such a pump is often called a **lifting pump.**

In order to prevent the leakage of water around the piston rod, a stuffingbox S is provided. The lower end of the discharge pipe P' is sometimes fitted with a valve c to prevent the water flowing back into the pump chamber; this valve is not essential to the operation of the pump, however, since the valve V prevents the water in the chamber and discharge pipe discharging during the downward motion of the piston.

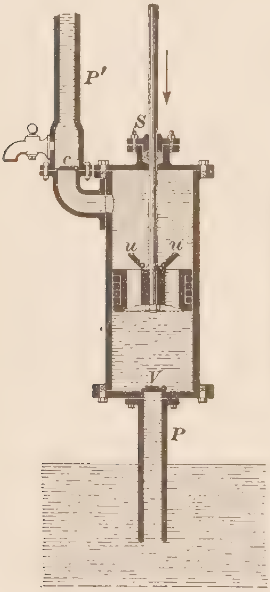


FIG. 26.

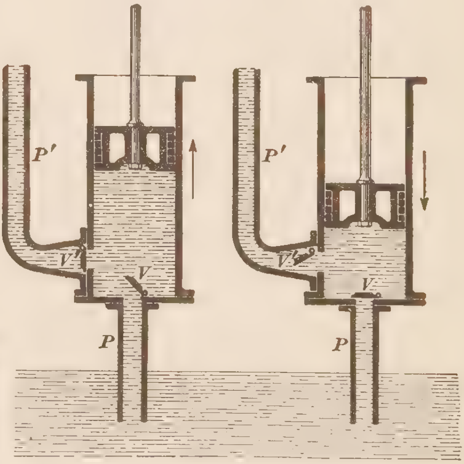


FIG. 27.

While it is customary to consider lifting pumps and suction pumps as two different types of pumps, there is in reality no difference in their operation, as a careful study of Figs. 25 and 26 will show. The only difference is that the water is discharged by a suction pump *at* the level of the pump, while a lifting pump discharges the water *above* the level of the pump.

73. Force Pumps.—The force pump differs from the lifting pump in one important particular, that is, in the fact that its piston is solid. A section of a force pump is shown in Fig. 27. As the piston ascends, as shown in the left-hand figure, the pressure of the atmosphere forces the water up the suction pipe P; the water opens the suction valve V and flows into the pump cylinder. When the piston moves down, as shown in the right-hand figure, the suction valve is closed and the delivery valve V' opened. The water in the pump cylinder is now forced up the delivery pipe P'. When the piston again begins to move upwards, the delivery valve is closed by the water above it and the suction valve opened by the pressure of the atmosphere on the water below it.

74. Plunger Pumps.—When force pumps are used to convey water to great heights or to force water against heavy pressures, the great pressure of the water makes it extremely difficult to keep the water from leaking past the piston, and the constant repairing and renewal of the piston packing becomes a nuisance and involves serious expense. To obviate this drawback, plunger pumps have been designed, one of which is shown diagrammatically in Fig. 28. The action

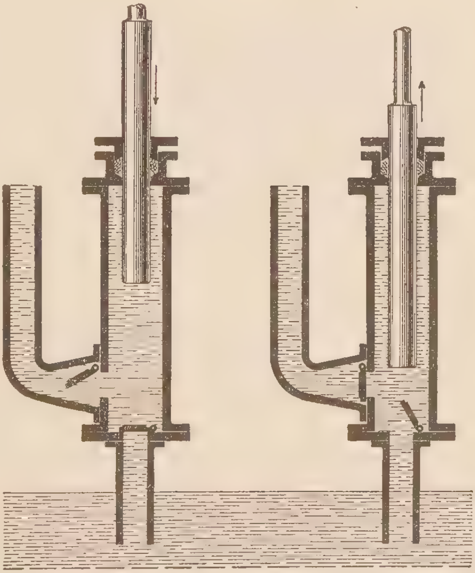


FIG. 28.

does not differ in any way from that of the piston force pump. During the up stroke of the plunger, the suction valve is open and the delivery valve is closed; during the down stroke, the suction valve is closed and the delivery valve is open.

75. The force pumps shown so far are **single-acting,** that is, the water is forced into the delivery pipe only during the downstroke or forward stroke of the piston or plunger. Force pumps, either of the piston or plunger pattern, may be constructed so as to force water into the delivery pipe both during the forward and return stroke. They are then called **double-acting.**

76. A **double-acting force pump** of the piston pattern is shown in Fig. 29. Such a pump has two sets of suction valves and delivery valves, one set for each side of the piston. With the piston moving in the direction of the arrow, the pressure of the atmosphere forces the water up the suction pipe *P* into the left-hand end of the pump cylinder, the left-hand suction valve opens and the left-hand delivery valve is closed. The piston in moving to the right, displaces the water in the right-hand end of the pump cylinder; as a consequence the right-hand suction valve is closed and the right-hand delivery valve opens. The water now flows up the delivery pipe *P'*. Imagine that the piston is at the end of its stroke and commences to move to the left. Its first movement

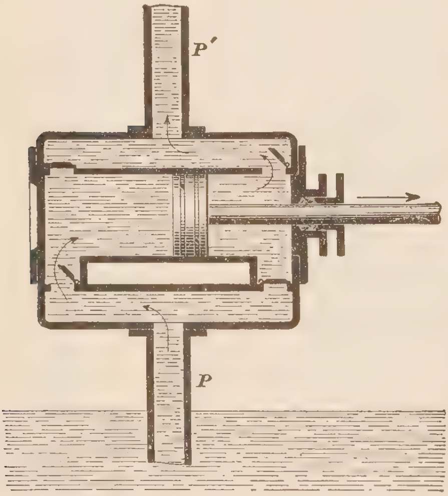


FIG. 29.

promptly closes the left-hand suction valve and opens the left-hand delivery valve. It also closes the right-hand delivery valve and opens the right-hand suction valve.

It is thus seen that with the arrangement given, which shows the principle of operation of all double-acting pumps, the piston will discharge water both during the forward and the return stroke. While the pump shown is a horizontal pump, it may be vertical as well.

STRENGTH OF MATERIALS.

GENERAL PRINCIPLES.

1. When a force is applied to a body, it changes either its form or its volume. A force, when considered with reference to the internal changes it tends to produce in any solid, is called a **stress.**

Thus, if we suspend a weight of 2 tons by a rod, the stress in the rod is 2 tons. This stress is accompanied by a lengthening of the rod, which increases until the internal stress or resistance is in equilibrium with the external weight.

2. Classification of Stresses.—Stresses may be classified as follows: *Tensile*, or *pulling stress; transverse*, or *bending stress; compressive*, or *pushing stress; shearing*, or *cutting stress; torsional*, or *twisting stress.*

3. A **unit stress** is the amount of stress on a unit of area, and may be expressed either in pounds or tons per square inch or per square foot; or, it is the load per square inch or per square foot on any body.

Thus, if 10 tons are suspended by a wrought-iron bar that has an area of 5 square inches, the unit stress is 2 tons per square inch, because $\frac{10}{5} = 2$ tons.

4. Strain is the deformation or change of shape of a body resulting from stress.

For example, if a rod 100 feet long is pulled in the direction of its length, and if it is lengthened 1 foot, it is strained $\frac{1}{100}$ its length, or 1 per cent.

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5. Elasticity is the property by virtue of which a body regains its original form after the external force on it is withdrawn, provided the stress has not exceeded the elastic limit. It is a property possessed by all bodies.

Consequently, we see from this that all material is lengthened or shortened when subjected to either tensile or compressive stress, and the change of the length within the elastic limit is directly proportional to the stress.

For stresses within the elastic limits, materials are perfectly elastic, and return to their original length on removal of the stresses; but when their elastic limits are exceeded, the changes of their lengths are no longer regular, and a permanent *set* takes place. The destruction of the material has then begun.

6. The **measure of elasticity** of any material is the change of length under stress within the elastic limit.

7. The **elastic limit** is that unit stress under which the permanent set becomes visible.

8. The elasticity of wrought iron and of all grades of steel is practically the same; that is, *within the elastic limit* each material will change an equal amount of length under the same stress. The elastic limit, however, is not the same for steel as for iron; it is higher for soft steel than for wrought iron, and, in general, the harder and stronger the steel the higher will be its elastic limit. As a consequence, steel will lengthen or shorten more than wrought iron, and hard steel more than soft, before its elasticity or ability to return to its original dimensions is injured.

TENSILE STRENGTH OF MATERIALS.

9. The **tensile strength** of any material is the *resistance* offered by its fibers to being pulled apart.

10. The *tensile strength* of any material *is proportional to the area of its cross-section.*

Consequently, when it is required to find the safe tensile strength of any material, we have only to find the area at the minimum cross-section of the body, and multiply it by the load per square inch that it can safely carry, as given in the following table under the heading "Safe Loads."

NOTE.—The minimum cross-section referred to in the above paragraph is that section of the material which is pierced with holes, such as bolt or rivet holes in iron, or knots in wood, if there are any.

TABLE I.

TENSILE STRENGTH OF MATERIALS.

Material.	Ultimate Tensile Strength. Pounds per Square Inch.	Safe Loads. Pounds per Square Inch.		
		Sudden.	Gradual.	Steady.
Timber	10,000	600	900	1,200
Cast iron	16,000– 20,000	1,600	2,400	3,200
Gun-metal cast iron	30,000– 35,000	2,000	3,000	4,000
Wrought iron	45,000– 55,000	4,500	9,000	13,000
Extra soft steel	45,000– 55,000	6,000	10,000	14,000
Flange steel	52,000– 62,000	6,000	10,000	14,000
Firebox steel	52,000– 62,000	6,000	10,000	14,000
Machinery steel	60,000– 75,000	6,500	11,000	15,000
Axle steel	75,000– 90,000	7,000	14,000	20,000
Hard steel (rail steel)	90,000–115,000	7,000	14,000	20,000
Tire steel	100,000–125,000	8,000	15,000	20,000
Crucible (tool) steel	125,000–180,000	15,000	23,000	30,000
Brass, cast	14,000– 20,000	1,400	2,200	3,000
Bronze, cast	30,000– 36,000	3,000	5,000	7,000
Tobin bronze	70,000– 80,000	7,000	11,000	14,000
Hard-drawn brass wire	30,000– 40,000	2,500	3,700	5,000
Copper, rolled	30,000– 36,000	3,500	6,000	8,000

11. In metals, a high tensile strength in itself is no indication of the ability of the metal to stand repeated applications of sudden stresses, as a high tensile strength usually involves a lesser degree of ductility than is obtained in metals of lower tensile strength. Steel of a tensile strength higher than 75,000 pounds is seldom used merely on account of its superior strength, but rather on account of its hardness, which enables it to better withstand abrasion.

12. The loads given in Table I are conservative safe loads deducted from experience and observation. They are given for material free from welds for such material as can be welded. While it is possible to make a weld as strong as the solid bar, the chances of the weld being imperfect are so great that it is unsafe to rely on such a degree of strength in welded bars. Furthermore, the value of the weld is an uncertain quantity that cannot be determined by an ocular inspection or any other inspection short of actually pulling the weld apart in a testing machine. Hence, it is advisable to reduce the safe loads given in the above table by 25 per cent. when a welded bar is subjected to a tensile stress. When judging the safe load of timber, due allowance must be made for knot holes and sappy spots.

13. It is often rather hard to determine whether a stress is steady, gradually varying, or gradually applied, or sudden. For example, considering the shell of a boiler, it would appear on first thought as though the tensile stress in the shell plates was a steady stress. But looking further into the problem, it will become apparent that the stress varies with the steam pressure, which gradually fluctuates within narrow or wider limits. Hence, most designers would consider the stress in a boiler shell as a gradually applied stress. In a piston rod or connecting-rod the load is applied almost instantly as soon as the crank passes over the center, and, hence, the stress would be considered to be a suddenly applied stress. When in doubt, it is good policy to err on the side of safety, that is, to choose a smaller safe load per square inch of section.

For special work, experience has indicated safe loads for different materials that fall below, or are above, those given in the table. Examples of this will be given later on.

14. The safe load per square inch of section is often called the **working stress per square inch,** or the **working load per square inch.** Care should be taken not to confound these terms with *working load*, *working stress*, *safe load*, or *safe tensile strength*, which, when applied without the limitation as to the unit of area, refer to the safe load the whole bar can carry.

RULES AND FORMULAS FOR TENSILE STRENGTH.

15. Let W = safe load in pounds;
A = area of minimum cross-section;
S = working stress in pounds per square inch, as given in Table I.

Rule 1.—*The working load in pounds for any bar subjected to a tensile stress is equal to the minimum sectional area of the bar, multiplied by the working stress in pounds per square inch, as given in the table.*

Or,
$$W = A S.$$

EXAMPLE.—A bar of good wrought iron that is 3 inches square is to be subjected to a steady tensile stress; what is the maximum load that it should carry?

SOLUTION.—According to the table, a working stress of 13,000 pounds may be allowed. Applying the rule, we have

$$W = 3 \times 3 \times 13{,}000 = 117{,}000 \text{ lb.} \quad \text{Ans.}$$

Rule 2.—*The minimum sectional area of any bar subjected to a tensile stress should be equal to the load in pounds, divided by the working stress in pounds per square inch, as given in the table.*

Or,
$$A = \frac{W}{S}.$$

Example.—What should be the area of a machinery-steel bar to carry a steady load of 108,000 pounds?

Solution.—According to the table, a safe load of 15,000 pounds per square inch of section may be allowed. Then, applying the rule,

$$A = \frac{108,000}{15,000} = 7.2 \text{ sq. in. Ans.}$$

Rule 3.—*The working stress in pounds per square inch is equal to the load in pounds divided by the minimum sectional area of the bar.*

Or,

$$S = \frac{W}{A}.$$

Example.—A piston rod 3 inches in diameter carries a piston 20 inches in diameter. With a steam pressure of 100 pounds to the square inch, what is the stress per square inch of section of the rod?

Solution.—The load on the piston is $20^2 \times .7854 \times 100 = 31,416$ pounds. The area of the rod is $3^2 \times .7854 = 7.0686$ square inches. Then, applying the rule just given, we have

$$S = \frac{31,416}{7.0686} = 4,444.4 \text{ lb. Ans.}$$

CHAINS.

16. **Chains** made of the same size iron vary in strength, owing to the different kinds of links from which they are made. It is a good practice to anneal old chains that have become brittle by overstraining. This renders them less liable to snap from sudden jerks. It reduces their tensile strength, but increases their toughness and ductility, which are sometimes more important qualities.

When annealing, care should be taken that a sufficient heat be applied, otherwise no benefit will be gained; the chains ought to be heated to a cherry red, say 1,300° F., at least.

17. The loads that can safely be lifted with chains are given in the following table. The safe load given is one-quarter of the steady load at which they may be expected to break. The table has been deduced from one published by the Lukens Iron and Steel Company.

TABLE II.

STRENGTH OF CHAINS.

Diameter of Rod of Which Link Is Made.	Safe Load. Pounds.	Diameter of Rod of Which Link Is Made.	Safe Load. Pounds.
$\frac{3}{16}$	430	1	12,500
$\frac{1}{4}$	770	$1\frac{1}{8}$	15,000
$\frac{5}{16}$	1,200	$1\frac{1}{4}$	18,000
$\frac{3}{8}$	1,750	$1\frac{3}{8}$	22,000
$\frac{7}{16}$	2,350	$1\frac{1}{2}$	26,000
$\frac{1}{2}$	3,100	$1\frac{5}{8}$	31,000
$\frac{9}{16}$	4,000	$1\frac{3}{4}$	36,000
$\frac{5}{8}$	4,800	$1\frac{7}{8}$	41,000
$\frac{11}{16}$	5,800	2	47,000
$\frac{3}{4}$	6,900	$2\frac{1}{4}$	56,000
$\frac{13}{16}$	8,000	$2\frac{1}{2}$	69,000
$\frac{7}{8}$	9,400	$2\frac{3}{4}$	84,000
$\frac{15}{16}$	11,000	3	100,000

STRENGTH OF ROPES.

HEMP ROPES.

18. The strength of hemp ropes does not depend entirely on the quality of the material and the cross-section of the rope, but to a large extent on the method of manufacture and the amount of twisting. A hemp rope is made up of fibers twisted together to form a **yarn,** several of which are laid up left-handed into **strands,** which in turn are twisted up right-handed into what is known as **plain-laid rope.**

Cable-laid or **hawser-laid rope** is left-handed rope of nine (9) strands.

Shroud-laid rope is formed by adding another strand to a plain-laid rope. Since the four strands on application of strain would sink in and detract from the rope's strength by an unequal distribution of strain, they are laid up around a **heart,** which is a small rope, made soft and elastic, and about one-third the size of the strands.

19. Ordinary hemp rope is designated by its circumference; rope for transmission purposes, which is either iron, steel, or manila rope, is designated by its diameter. It is well to keep this in mind. The loads that can safely be hoisted with hemp ropes of different sizes is given in the following table prepared by Ensign F. R. Brainard, United States Navy.

TABLE III.

STRENGTH OF HEMP ROPES.

Circumference. Inches.	Safe Load. Pounds.	Circumference. Inches.	Safe Load. Pounds.
1.00	65	5.75	2,680
1.25	125	6.00	2,975
1.50	185	6.50	3,470
1.75	255	7.00	3,965
2.00	320	7.50	4,570
2.25	410	8.00	5,175
2.50	500	8.50	5,910
2.75	610	9.00	6,640
3.00	715	9.50	7,370
3.25	850	10.00	8,105
3.50	980	10.50	8,970
3.75	1,135	11.00	9,840
4.00	1,285	11.50	11,015
4.25	1,450	12.00	12,190
4.50	1,610	12.50	13,365
4.75	1,750	13.00	14,540
5.00	1,990	13.50	15,845
5.25	2,190	14.00	17,150
5.50	2,385	14.50	18,450

20. The above table applies to new ropes or ropes in first-class condition. If the rope is used for a block and tackle, the bending of the rope around the pulleys will cause a rapid deterioration of the inside, owing to the chafing and sliding of the strands and yarns upon one another. The outside appearance of a rope used for block and tackle is in itself no indication of its quality; ropes are frequently found that appear perfectly sound when judged by their outside appearance and yet are entirely unsafe. To inspect the rope, take it in both hands and untwist it sufficiently to expose the inner surfaces that have been chafing against one another. If a considerable number of broken fibers are found, the safe load should be reduced by 50 per cent. If a rather large quantity of the fibers have been reduced to powder, the rope should be condemned and preferably destroyed immediately, in order to remove any temptation to use it. A rope in this condition is liable to give way suddenly under a very light load.

Ropes used for slings and lashings, as a general rule, are ruined by the external chafing they receive and, hence, their safety can be determined by their outward appearance.

21. Slings are chiefly used for attaching tackles to machine parts during the erection and repair of machinery. They are made by taking a piece of rope a little more than twice the required length of the rope and joining the ends by splicing, the splice known as a **short splice** being usually employed. It should be distinctly understood that this splice is not intended nor adapted for ropes that are to pass over a pulley.

22. Splicing a Sling.—To make a short splice in a sling, untwist the strands for some distance from each end of the rope. Take an end in each hand and lay the strands within one another, as shown in Fig. 1 (*a*). The strands *1*′, *2*′, and *3*′ may be tied to the left-hand end for convenience in handling. Now take the strand *1*, pass it over strand *2*′ and under strand *1*′, as shown in Fig. 1 (*b*); then, draw it tight. Next, take strand *2*, pass it over *3′* and under *2′*, drawing it up tightly. Now take strand *3*, pass it over *1′* and under *3′*; then draw it up. The splice will now appear as shown in Fig. 1 (*c*). Weave the strands *1′*, *2′*, and *3′* into the left-hand rope in the same manner. Continue weaving the right-hand and left-hand strands alternately until the length of the splice is about 6 times the diameter of the rope. Draw the strands tight and subject the splice to a strain. Then cut off the projecting ends of the strands.

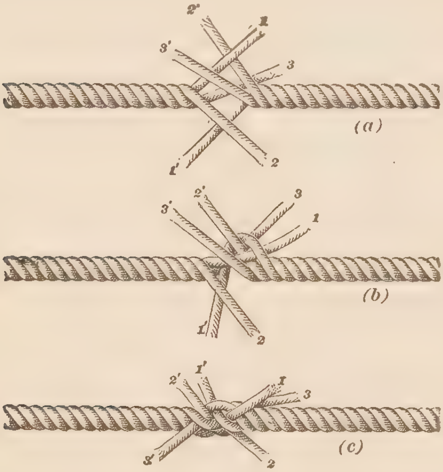


FIG. 1.

23. Use of Slings.—The weight that can safely be lifted by a sling depends not only on the size of the rope of which it is made, but also on the manner in which it is attached to the hook of the tackle block. Thus, in Fig. 2, the sling is simply hooked over the hook.

Owing to the sharp bend over the hook, the strength of the sling is greatly reduced, and the direct load for this manner of attachment should not exceed the safe load given in the table for one rope of equal circumference, although the load is borne by two parts of the rope.

When the sling is attached to the hook by doubling over, as shown in Fig. 3, its greatest strength is realized, and for the two parts of the rope parallel to each other, the stress in the rope will be equal to half the load, as the load is supported by two ropes. Then, with a sling doubled over as shown, a load twice as great can safely be hoisted than when applied as shown in Fig. 2.

In the foregoing statement, it is assumed that the sling is fastened to the work without a sharp bend, the equivalent of a single loop over a hook. The stress in the two parts of a sling will be least when they are parallel to each other; hence, a sling will then lift the greatest load with safety. The stress becomes greater when the angle between the two parts of the sling is increased; its magnitude for any angle can be determined by the method of the triangle of forces.

24. A quick way of attaching a rope to a hook is that shown in Fig. 4, and known as a **Blackwall hitch.** When making this hitch, always make it as shown in the figure; that is, let the detached end lie between the hook and the end sustaining the load.

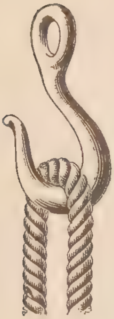


FIG. 2.

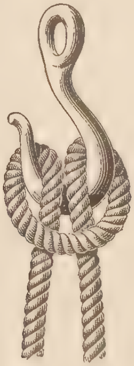


FIG. 3.

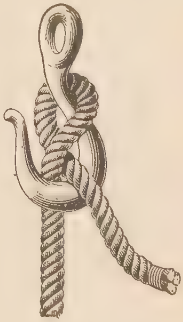


FIG. 4.

25. The proper size of a rope for a block and tackle is fixed by the size of the groove turned in the pulleys. Then, from the table, take the safe working load of the rope, and using this as the force applied to the free end of the rope, the maximum load a given block and tackle can lift safely can be calculated by the rule given in connection with the subject of pulleys.

MANILA ROPE.

26. Manila rope is used chiefly for hoisting purposes and for power transmission. The so-called "Stevedore" manila rope has the fibers lubricated with a mixture of tallow and plumbago during the process of manufacture. This lubrication prevents internal chafing and wear to a large extent.

Manila rope for hoisting and power-transmission purposes is usually made with four strands, that is, shroud laid. Hoisting rope is ordered by *circumference;* transmission rope by *diameter.*

27. The working stress on manila hoisting rope should not exceed that given in the table below, which was published by C. W. Hunt & Company.

TABLE IV.

STRENGTH OF MANILA HOISTING ROPE.

Circumference of Rope. Inches.	Working Stress.
3	350
3½	500
4	650
4½	800
5	1,000

Under ordinary conditions, with a working stress not exceeding that given in the table, a manila hoisting rope will last until a total load of about 6,000 tons has been hoisted with it. Under exceptionally favorable circumstances, where the rope runs over large sheaves, a manila rope has hoisted 20,000 tons. Manila ropes for hoisting purposes should not be spliced, as it is difficult to make a splice that will not pull out while running over the sheaves, and the increased wear to be obtained from splicing a broken hoisting rope is usually very small.

28. Manila rope for power transmission is made from $\frac{1}{2}$ inch up to 2 inches in diameter. The greatest horsepower can be transmitted by a rope when the rope speed is about 5,000 feet per minute. When run at greater speed, the effect of the centrifugal force in the rope rapidly decreases the horsepower that can be transmitted. The size of the pulley has an important bearing on the wear of the rope; in general, it should not be smaller than 40 times the diameter of the rope, and as much larger as it can conveniently be made.

WIRE ROPES.

29. Wire rope is made of iron and steel wire. It is stronger than hemp rope, and, to carry the same load, is of smaller diameter.

In substituting steel for iron rope, the object in view should be to gain an increase of wear from the rope, rather than to reduce the size. A steel rope to be serviceable should be of the best obtainable quality, because ropes made from low grades of steel are inferior to good iron ropes.

30. Rules for the strength of wire ropes:

Let W = maximum of working load in pounds;
C = circumference of rope in inches.

Rule 4.—*The maximum working load in pounds that should be allowed on any iron-wire rope is equal to the square of the circumference of the rope in inches, multiplied by 600.*

Or, $$W = 600\,C^2.$$

Example.—What is the maximum load in pounds that should be carried by an iron rope whose circumference is $4\frac{1}{2}$ inches?

Solution.—Applying the rule just given, we have

$$W = 600 \times 4.5^2 = 12{,}150 \text{ lb.} \quad \text{Ans.}$$

Rule 5.—*The circumference of any iron-wire rope in inches is equal to the square root of the maximum working load in pounds, multiplied by .0408.*

Or, $$C = .0408\sqrt{W}.$$

Example.—A maximum working load of 12,150 pounds is to be carried by an iron-wire rope; what should be the minimum circumference of the rope?

Solution.—Applying rule **5**, we have

$$C = .0408\sqrt{12{,}150} = 4\tfrac{1}{2} \text{ in.} \quad \text{Ans.}$$

Rule 6.—*The above rules and formulas are also made applicable when computing the safe strength of steel-wire rope by substituting the constant 1,000 for the constant 600, and .0316 for .0408.*

Example.—What is the maximum load in pounds that should be carried by a steel-wire rope, the circumference of which is 4½ inches?

Solution.—Applying rule **4**, we have

$$W = 1{,}000 \times 4.5^2 = 20{,}250 \text{ lb.} \quad \text{Ans.}$$

Example.—A maximum working load of 10,485 pounds is to be carried by a steel-wire rope; what should be the minimum circumference of the rope?

Solution.—Applying rule **5**, we have

$$C = .0316\sqrt{10{,}485} = 3.24 \text{ in.} \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. What should be the diameter of a machinery-steel piston rod of a steam engine to resist tension, if the piston is 19 inches in diameter and the pressure is 85 pounds per square inch? Ans. 2¼ in., nearly.

2. What safe load will a cast-iron bar of rectangular cross-section 7½ in. by 3½ in. support if subjected to shocks? The bar is in tension. Ans. 42,000 lb.

3. What is the stress per square inch on a piece of timber 8 inches square that is subjected to a steady pull of 60,000 pounds? Ans. 937.5 lb. per sq. in.

4. What should be the allowable working load for a steel-wire rope whose circumference is 3¾ inches? Ans. 14,062.5 lb.

5. What should be the circumference of an iron-wire rope to support a load of 20,000 pounds? Ans. 5¾ in., nearly.

CRUSHING STRENGTH OF MATERIALS.

31. The **crushing strength** of any material is the resistance offered by its fibers to being pushed together.

32. To obtain only compression, the length of a rod should not be more than five times greater than its least diameter or its least thickness when it is a rectangular rod. Such a rod is called a **short column.**

If a bar is long compared with its cross dimensions, the load, if sufficiently great, will cause it to bend sidewise under the compressive force, and we have, then, not only compression, but compression compounded with bending.

33. When a bar or rod is longer than five times its least diameter or least thickness, it is called a **long column,** or **long pillar.**

34. The shape of the *ends* of a column has great influence on its strength. In Fig. 5 are shown three columns with differently shaped ends.

It has been proved by the aid of higher mathematics that, theoretically, a pillar, as *a*, having flat or fixed ends, is 4 times as strong as one that has round or movable ends, as the pillar *c*, and $1\frac{7}{9}$ times as strong as one having one flat and one round end, as the pillar *b*; *b* is thus $2\frac{1}{4}$ times as strong as *c*. It has also been found that if three pillars *a*, *b*, *c*, which have the same cross-section, are to carry the same load and be of equal strength, their lengths must be as the numbers 2, $1\frac{1}{2}$, and 1, respectively.

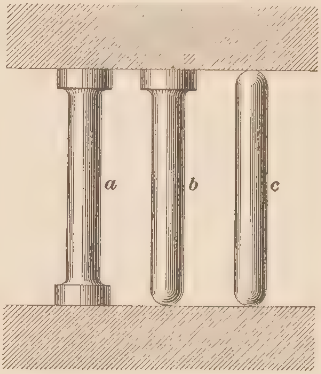


FIG. 5.

In practice, however, the ends of the pillars *b* and *c* are not generally made as shown by the figure, but have holes at their ends into which pins are fitted that are fastened to some other piece. In such cases, it has been found that *a* is

2 times as strong as *c* and that *b* is 1½ times as strong as *c*. That is, in actual practice, a column fixed as at *c* is really ½ as strong as one fixed as at *a*, instead of being only ¼ as strong, as given above.

An example of a column fixed at both ends is the piston rod of a steam engine. The valve stem of a slide-valve engine may be considered as an example of a column fixed at one end and movable at the other. The connecting-rod of a steam engine is a good example of a column having two movable ends.

35. Average values of the crushing strength of various materials used in engineering are given in the table below.

TABLE V.

CRUSHING STRENGTH OF MATERIALS.

Material.	Crushing Strength. Pounds per Sq. In.	Safe Loads. Pounds.		
		Sudden.	Gradual.	Steady.
Cast iron	80,000	7,000	11,000	14,000
Wrought iron	45,000	3,500	4,500	6,000
Machinery steel	60,000	4,000	6,000	8,000
Cast brass	9,000	800	1,200	1,500
Timber, dry	7,000	600	850	1,100
Timber, wet	3,000	250	380	500
Stone	6,000	....	....	200
Brick	2,000	....	....	100

STRENGTH OF COLUMNS.

36. In practice, columns subjected to a compressive stress are made of cast iron, wrought iron, steel, or timber. For short columns, the area, etc. can be calculated by the same rules that have been given for tensile stresses, substituting the working loads per square inch given in the above table.

37. The **safe working load on long columns** is given by the following rule, which is applicable to columns the length of which does not exceed 40 times their least diameter or their least thickness when rectangular. The rule given applies to columns uniform throughout their length. The rules for tapering columns involve considerable mathematical knowledge and are so rarely used in practice that they will not be given here.

Let C = safe crushing load, as given in the table;
S = sectional area in square inches;
L = length of column in inches;
d = least thickness of rectangular column, or diameter of round column, or dimension indicated by the arrowheads in the following table, in inches;
W = safe working load in pounds;
A = area of the two flanges in square inches;
B = area of the web in square inches;
a = constant corresponding to the cross-section of the column, as given in Tables VI, VII, and VIII.

Rule 7.—*The safe working load of a long column, in pounds, is equal to the safe working load corresponding to the kind of stress given in Table V multiplied by the sectional area of the column, in square inches, and the product divided by 1 plus the quotient obtained by dividing the square of the length of the pillar in inches by the square of the diameter* (*or least thickness, if rectangular*) *multiplied by the value of a.*

Or,
$$W = \frac{C\,S}{1 + \frac{L^2}{a\,d^2}}.$$

38. When using this formula, first obtain the value of C from Table V. Next, calculate the area of the cross-section of the pillar. Then find the value of a from one of the last three tables. Finally, be sure that the length of the column has been reduced to inches before substituting in the formula.

TABLE VI.

CONSTANTS FOR WROUGHT-IRON AND STRUCTURAL-STEEL PILLARS.

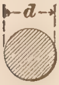
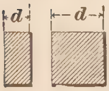
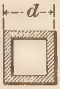
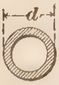
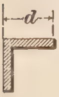
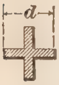
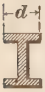
Cross-Section of Pillar.	When Both Ends of the Pillar Are Flat or Fixed.	When One End of the Pillar Is Flat or Fixed and the Other Round or Movable.	When Both Ends of the Pillar Are Round or Movable.
 Round.	2,250	1,500	1,125
 Square or Rectangle.	3,000	2,000	1,500
 Thin Square Tube.	6,000	4,000	3,000
 Thin Round Tube.	4,500	3,000	2,250
 Angle With Equal Sides.	1,500	1,000	750
 Cross With Equal Arms.	1,500	1,000	750
 I Beam.	$3{,}000 \times \frac{A}{A+B}$	$2{,}000 \times \frac{A}{A+B}$	$1{,}500 \times \frac{A}{A+B}$

TABLE VII.

CONSTANTS FOR CAST-IRON PILLARS.

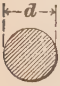
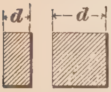
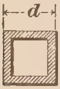
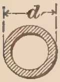
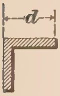
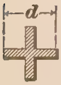
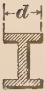
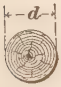
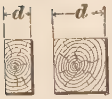
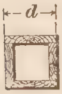
Cross-Section of Pillar.		When Both Ends of the Pillar Are Flat or Fixed.	When One End of the Pillar Is Flat or Fixed and the Other Round or Movable.	When Both Ends of the Pillar Are Round or Movable.
	Round.	281.25	187.5	140.625
	Square or Rectangle.	375.00	250.0	187.500
	Thin Square Tube.	750.00	500.0	375.000
	Thin Round Tube.	562.50	375.0	281.250
	Angle With Equal Sides.	187.50	125.0	93.750
	Cross With Equal Arms.	187.50	125.0	93.750
	I Beam.	$375 \times \frac{A}{A+B}$	$250 \times \frac{A}{A+B}$	$125 \times \frac{A}{A+B}$

TABLE VIII.

CONSTANTS FOR WOODEN PILLARS.

Cross-Section of Pillar.	When Both Ends of the Pillar Are Flat or Fixed.	When One End of the Pillar Is Flat or Fixed and the Other Round or Movable.	When Both Ends of the Pillar Are Round or Movable.
 Round.	187.5	125.00	93.75
 Square or Rectangle.	250.0	166.66	125.00
 Hollow Square Made of Boards.	500.0	333.33	250.00

To find the proper value of a in any example, first turn to the table dealing with the material in question, and find the figure corresponding to the given cross-section; in the horizontal line containing this are three numbers corresponding to the different conditions of the ends of the column. From these numbers select the one corresponding to the given conditions of the column to be calculated, and this will be the required value of a.

NOTE.—If the length of the pillar is given in feet, be sure to reduce it to inches before substituting in the formula.

39. Rule 7 cannot be transformed to determine directly the diameter or area of a column of a given cross-section required to sustain a stated load. An area and a corresponding value of d must be assumed and the safe load corresponding to these values calculated. If the safe load is smaller than the load to be sustained, a larger area and value of d must be chosen. If the calculated safe load is larger, a somewhat smaller area and value of d may be tried.

When applying the rule given to a column subject to wear, as, for instance, the piston rod of an engine, the diameter of the rod should be increased somewhat over that given by calculation in order to allow for subsequent truing up. This allowance may be from $\frac{1}{8}$ to $\frac{1}{4}$ of an inch, according to the judgment of the designer.

40. A rough-and-ready rule for the size of a piston rod is to make it one-sixth the diameter of the cylinder of a simple engine. This rule of thumb allows us to choose a value of S and d that will answer very well indeed for the first trial.

Illustration.—What size machinery-steel piston rod is required for a simple engine having a piston 36 inches in diameter to carry a steam pressure (gauge) of 100 pounds per square inch; the rod is 70 inches long?

According to the rule given, an approximate size of the piston rod will be $36 \div 6 = 6$ inches. The area corresponding to this is $6^2 \times .7854 = 28.27$ square inches. From Table VI, $a = 2{,}250$. The piston rod being subjected to suddenly applied stresses, by Table V, the allowable safe working load per square inch is 4,000 pounds. The working load on the piston rod is $36^2 \times .7854 \times 100 = 101{,}787.8$ pounds. Applying rule **7** and substituting values, we get

$$W = \frac{4{,}000 \times 28.27}{1 + \frac{70^2}{2{,}250 \times 6^2}} = 106{,}629 \text{ pounds,}$$

which is a safe load on a 6-inch piston rod under the given conditions. This tends to show that the rod is rather large.

Now try a $5\frac{7}{8}$-inch rod. The area of the rod is $5\frac{7}{8}^2 \times .7854 = 27.1$ square inches, nearly.

Applying rule **7** again and substituting the new values of S and d, we get

$$W = \frac{4{,}000 \times 27.1}{1 + \frac{70^2}{2{,}250 \times 5\frac{7}{8}^2}} = 102{,}264 \text{ pounds, nearly.}$$

It is thus seen that a 5⅞-inch rod is just about right. Most designers would allow ⅛ inch for truing up, thus making the rod 6 inches in diameter.

In this particular case the size of rod found by calculation agrees with that given by the rough-and-ready rule. This cannot be in every case, however, as will readily be seen if a steam pressure of 150 pounds be substituted for 100 pounds and the rod be calculated for this pressure. It will then be found that a rod larger than 6 inches will be required.

41. In calculating the size of piston rods in actual work, it is good practice to assume at least a steam pressure of 100 pounds for the calculation, even though it is intended to carry a lower pressure. An extra margin of strength is thus provided which may be needed in the future, as cases are very frequent where engines designed for 75 pounds have ultimately been run at 100 pounds pressure.

42. In engineers' examinations, candidates for license are often asked to calculate the size of the piston rod for a given engine. A rule that is easily remembered is the following, which assumes the piston rod to be a short column and makes allowance for failure by bending by reducing the allowable working stress given in the table by 10 per cent., thus making the safe working stress for steel piston rods 3,600 pounds per square inch and, say, 3,100 pounds for iron.

This rule will give fairly good results within the limits of ordinary practice and will usually satisfy an examiner. For an actual design, however, rule **7**, which is more rational, is preferable.

Let S = area of piston rod;
l = load in pounds on the rod = area of piston × steam pressure;
c = safe working load in pounds.

Rule 8.—*Multiply the area of the piston by the steam pressure and divide the product by the safe working load corresponding to the material. The quotient will be the required area.*

Or, $$S = \frac{l}{c}.$$

Example.—What size steel piston rod is required for a 36-inch cylinder with a steam pressure of 100 pounds?

Solution.—The safe working load to be used in applying rule 8 is 3,600 pounds.

Then, $$S = \frac{36^2 \times .7854 \times 100}{3,600} = 28.27 \text{ square inches.}$$

The corresponding diameter is $\sqrt{\frac{28.27}{.7854}} = 6$ in. Ans.

EXAMPLES FOR PRACTICE.

1. A round wrought-iron column 4 inches in diameter and 60 inches long is subjected to a steady load. The column is fixed at one end and movable at the other. What load will it sustain?
Ans. 65,564 lb., nearly.

2. A solid machinery-steel column with both ends hinged is $4\frac{1}{2}$ inches in diameter and 10 feet long. It is subjected to a suddenly applied stress. What is the safe working load? Ans. 38,979 lb.

3. A rectangular wooden column is 14 feet long. One end is fixed and one end is movable. If the cross-section is 12 in. × 8 in., what is its safe load for a steady stress? Ans. 28,963 lb.

TRANSVERSE STRENGTH OF MATERIALS.

43. The **transverse strength** of any material is the resistance offered by its fibers to being broken by bending. As, for example, when a beam, bar, rod, etc. which is supported at its ends is broken by a force applied between the supports.

The transverse strength of any beam, bar, rod, etc. is proportional to the product of the square of its depth multiplied by its width; consequently, it is more economical to increase the depth than the width.

44. The safe load that can be carried by beams, bars, rods, etc. of uniform cross-section depends on the material, the manner in which the beam is supported and loaded, and on the shape of the cross-section. A beam that is rigidly fixed at one end and free at the other, when subjected to a transverse stress, is called a **cantilever.**

45. The safe working loads on beams supported and loaded in different ways can be found by applying the corresponding formula given in Table IX. In the formulas given, $W =$ the safe load in pounds, and $l =$ length of beam in inches to be taken as shown in the illustrations of Table IX. To apply the formulas to a beam, the values of S must be taken from Table X; and the value of R is to be computed by the formulas given in Table XI. It may then be substituted. Attention is called to the fact that it is absolutely necessary to reduce the length of the beam to inches; furthermore, the load W on a uniformly loaded beam is the total load for the length l, and not the load per unit of length.

In the table giving the value of R, the letter A denotes the whole area of the section in square inches, that is, taking the hollow cylinder, for example, it denotes the area corresponding to its outside diameter. The letter a stands for the area of the hollow part of the section. In the formulas where A is applied to a solid section, it stands for the area of the section.

The values of the constant S have been determined from practical experience and are safe, conservative values for the conditions stated.

Example 1.—A cast-iron, solid, rectangular beam 8 inches deep and 4 inches wide rests upon two supports 8 feet apart. What steady load will it safely support when the load is applied at the middle?

Solution.—According to Table IX, the formula to be used is $W = \frac{4 S R}{l}$. For a steady stress on cast iron, Table X gives 7,500 as

TABLE IX.

FORMULAS FOR STRENGTH OF BEAMS.

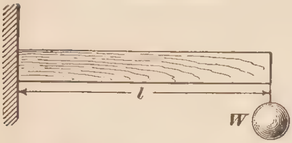
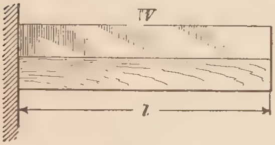
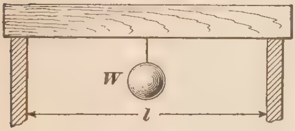
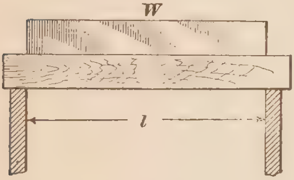
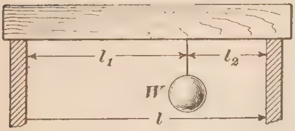
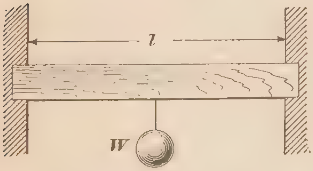
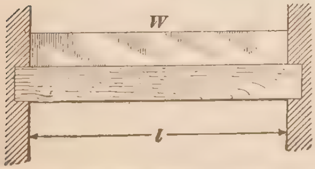
Manner of Supporting Beams.	Remarks.	Formula for Safe Load. Pounds.
	Cantilever load *W* at free end.	$W = \frac{SR}{l}$
	Cantilever, uniformly loaded. *W* = total load.	$W = \frac{2\,SR}{l}$
	Simple beam resting on two supports, load *W* at middle.	$W = \frac{4\,SR}{l}$
	Simple beam resting on two supports, uniformly loaded. *W* = total load on length *l*.	$W = \frac{8\,SR}{l}$
	Simple beam resting on two supports, single load *W* not in the middle.	$W = \frac{SR}{\frac{l_1 l_2}{l}}$
	Beam rigidly fixed at both ends, load *W* in middle.	$W = \frac{8\,SR}{l}$
	Beam rigidly fixed at both ends, uniformly loaded. *W* = total load on length *l*.	$W = \frac{12\,SR}{l}$

TABLE X.

VALUES OF S.

Material.	Nature of Load.		
	Suddenly Applied.	Gradually Applied.	Steady.
Cast iron	2,250	3,000	7,500
Wrought iron	4,000	6,000	13,700
Structural steel	5,000	7,500	16,800
Brass	1,100	1,500	3,600
White pine	320	480	960
Yellow pine	500	730	1,460
Hemlock	240	360	720
Oak	400	600	1,200

the value of S. For a solid rectangular beam, Table XI gives $R = \frac{b\,h^2}{6}$. Substituting the width and depth of the beam in this last formula, we get $R = \frac{4 \times 8^2}{6} = 42.67$, nearly. The length of the beam in inches is $8 \times 12 = 96$ inches. Substituting all the values in the formula for the safe working load, we get

$$W = \frac{4 \times 7{,}500 \times 42.67}{96} = 13{,}334 \text{ lb., nearly. Ans.}$$

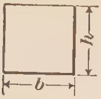
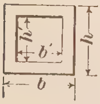
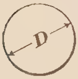
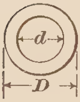
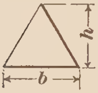
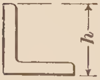
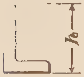
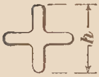
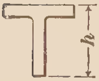
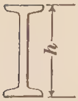
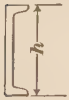
EXAMPLE 2.—An I beam having a depth of 10 inches and an area of section of 7.5 square inches is to be used as a cantilever to bear a varying load that is to be uniformly distributed. The beam is made of steel and is 100 inches long. What safe load will it carry?

SOLUTION.—From Table IX, it appears that the proper formula to be used is $W = \frac{2\,S\,R}{l}$. According to Table X, the value of S for steel for a gradually applied load is 7,500. By Table XI for an **I** beam, $R = \frac{A\,h}{3.33}$. Substituting values in this last formula, we get $R = \frac{7.5 \times 10}{3.33} = 22.52$, nearly. Substituting values in the formula for the safe working load, we get

$$W = \frac{2 \times 7{,}500 \times 22.52}{100} = 3{,}378 \text{ lb. Ans.}$$

TABLE XI.

VALUES OF *R*.

Section.	R.
	$\dfrac{b\,h^2}{6}$
	$\dfrac{b\,h^3 - b'\,h'^3}{6\,h}$
	$\dfrac{A\,D}{8}$
	$\dfrac{A\,D^2 - a\,d^2}{8\,D}$
	$\dfrac{b\,h^2}{24}$
	$\dfrac{A\ h}{7.2}$
	$\dfrac{A\ h}{6.5}$
	$\dfrac{A\ h}{9.5}$
	$\dfrac{A\ h}{8}$
	$\dfrac{A\ h}{3.33}$
	$\dfrac{A\ h}{3.67}$

46. While the formulas given in Table IX will allow the safe load on a given beam to be calculated, they cannot readily be transformed to give directly the size of beam required under given conditions to carry a given load.

In practice, when it is required to determine the size of beam, the length between supports is known. After the shape of cross-section of the beam has been chosen, dimensions for the beam may be assumed and its safe load for the given dimensions calculated. If the load thus found falls below the load the beam is to carry, larger dimensions must be chosen. In case of rolled sections, such as angle irons, **T** irons, channels, or **I** beams, catalogues of manufacturers should be consulted and standard sizes chosen. These catalogues usually contain the area of the section for different weights and dimensions of rolled sections.

47. The values of R given in Table XI *apply only when the dimension marked h is vertical.* If the beam is placed in any other position, R will have a different value, which can only be calculated by the aid of higher mathematics. Beams rigidly fixed at both ends are rarely met with in practice, and it is safer to assume that the beam merely rests on two supports.

EXAMPLES FOR PRACTICE.

1. What weight will a yellow-pine rectangular beam carry safely under a steady load when used as a cantilever and uniformly loaded? The beam is 9 feet 6 inches long, 16 inches deep, and 4 inches wide. Ans. 4,371 lb.

2. What weight would the beam in example 1 carry safely if it rested on two supports? Ans. 17,484 lb.

3. What weight can safely be carried at the end of a wrought-iron round cantilever 3 inches in diameter when the load is suddenly applied 4 feet 2 inches from the support of the cantilever? Ans. 212 lb.

4. What steady load can be carried safely by the same beam given in the preceding example? Ans. 726 lb

SHEARING, OR CUTTING, STRENGTH OF MATERIALS.

48. The **shearing strength** of any material is the resistance offered by its fibers to being cut in two. Thus, the pressure of the cutting edges of an ordinary shearing machine, Fig. 6, causes a shearing stress in the plane *a b*. The unit shearing force may be found by dividing the force *P* by the area of the plane *a b*.

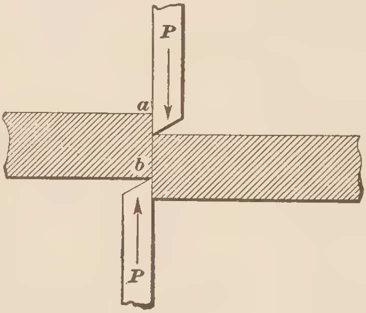


FIG. 6.

Fig. 7 shows a piece in double shear; here the central piece *c d* is forced out while the ends remain on their supports *M* and *N*.

The shearing strength of any body is directly proportional to its area.

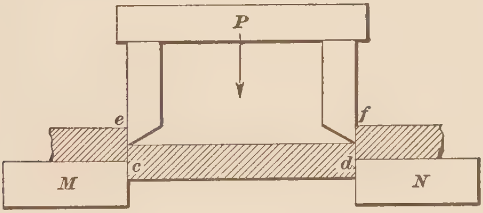


FIG. 7.

49. In Table XII are given the greatest and the safe shearing strengths per square inch of different kinds of materials.

50. Rules for Shearing.—In general, the force required to shear a piece of material in double shear will be *twice* that required for single shear. This does not apply to all cases, however; a notable exception are rivets in double shear. Experiments have determined that the force required to shear both iron and steel rivets in double shear averages about 1.85 times that required for single shear.

TABLE XII.

SHEARING STRENGTH OF MATERIALS.

Material.	Average Ultimate Shearing Strength.		Safe Loads.					
			Sudden Load.		Gradual Load.		Steady Load.	
	Parallel to Grain.	Across Grain.	Parallel to Grain.	Across Grain.	Parallel to Grain.	Across Grain.	Parallel to Grain.	Across Grain.
Cast iron	25,000	....	2,500	...	3,700	...	5,000	...
Wrought-iron rivets	38,000	....	3,700	...	5,600	...	7,500	...
Soft-steel rivets	44,000	....	4,400	...	6,600	...	8,800	...
Ordinary wrought iron	44,000	....	4,400	...	6,600	...	8,800	...
Structural steel	52,000	....	5,000	...	7,500	...	10,000	...
White pine	300	2,500	30	250	45	375	60	500
Yellow pine	400	4,500	40	450	60	675	80	900
Hemlock	250	2,500	25	250	38	375	50	500
Spruce	300	3,000	30	300	45	450	60	600
Oak	600	5,000	60	500	90	750	120	1,000

Let a = area of cross-section in square inches;
S = shearing stress as given in table;
W = load in pounds.

Then, to find the safe load or the load that will shear the material:

Rule 9.—*Multiply the area of the section by the shearing stress.*

Or, $$W = a\,S.$$

In applying this rule, it should be remembered that when the safe load the material can bear is required, the value of S is to be taken from the column headed "Safe Loads," selecting the safe load corresponding to the nature of the load. When it is desired to find the load at which the material will fail, the value of S is to be taken from the column headed "Ultimate Shearing Strength."

EXAMPLE.—If the beam in double shear shown in Fig. 7 is rectangular and measures 4 in. × 2 in., what steady safe load would you allow if the beam were made of structural steel?

SOLUTION.—According to the table, a safe load of 10,000 pounds per square inch of section may be allowed. Applying rule 9 and multiplying by 2, since the beam is in double shear, we get

$$W = 4 \times 2 \times 10{,}000 \times 2 = 160{,}000 \text{ lb.} \quad \text{Ans.}$$

51. To find the area required for a material subjected to a shearing stress:

Rule 10.—*Divide the load by the shearing stress.*

Or, $$a = \frac{W}{S}.$$

In applying rule **10**, it should be remembered that for double shear the result is to be divided by 2 to obtain the area of the beam.

EXAMPLE.—A white pine beam is subjected to a suddenly applied shearing stress by a load of 4,000 pounds, the beam being in double shear and the load being applied across the grain. What should be the area of the beam to bear this stress safely?

SOLUTION.—By the table, $S = 250$. Then, by rule **10,** we get

$$a = \frac{4{,}000}{250} = 16 \text{ square inches for single shear.}$$

Dividing the result by 2 for double shear, we get $16 \div 2 = 8$ sq. in. area. Ans.

EXAMPLES FOR PRACTICE.

1. A hemlock beam 8 in. × 2 in. is in single shear parallel to the grain. What load will cause it to fail by shearing? Ans. 4,000 lb.

2. A wrought-iron rivet 1 inch in diameter is in double shear. At what load will it be likely to shear? Ans. 55,214 lb., nearly.

TORSION.

52. When a force is applied to a beam, bar, or rod in such a manner that it tends to twist it, the stress thus produced is termed **torsion.** Torsion manifests itself in the case of rotating shafts, such as line shafts and engine shafts.

LINE SHAFTING.

53. A **line of shafting** is one continuous run, or length, composed of lengths of shafts joined together by couplings.

54. The **main line of shafting** is that which receives the power from the engine or motor and distributes it to the other lines of shafting or to the various machines to be driven.

Line shafting is supported by hangers, which are brackets provided with bearings, bolted either to the walls, posts, ceilings, or floors of the building. Short lengths of shafting, called **countershafts,** are provided to effect changes of speed and to enable the machinery to be stopped or started.

55. Shafting is usually made cylindrically true, either by a special rolling process, when it is known as **cold-rolled shafting,** or else it is turned up in a machine called a **lathe.** In the latter case it is called **bright shafting.** What is known as **black shafting** is simply bar iron rolled by the ordinary process and turned where it receives the couplings, pulleys, bearings, etc.

56. The diameter of bright turned shafting increases by $\frac{1}{4}$ inch up to about $3\frac{1}{2}$ inches in diameter; above this diameter it increases by $\frac{1}{2}$ inch. The actual diameter of a bright shaft is $\frac{1}{16}$ inch less than the commercial diameter, it being designated from the diameter of the ordinary round bar iron from which it is turned. Thus, a length of what is called 3-inch bright shafting is really only $2\frac{15}{16}$ inches in diameter.

Cold-rolled shafting is designated by its commercial diameter; thus, a length of what is called 3-inch shafting is 3 inches in diameter.

57. In the following table is given the maximum distance between the bearings of some continuous shafts that are used for the transmission of power:

TABLE XIII.

DISTANCE BETWEEN BEARINGS.

Diameter of Shaft. Inches.	Distance Between Bearings. Feet.	
	Wrought-Iron Shaft.	Steel Shaft.
2	11	11.50
3	13	13.75
4	15	15.75
5	17	18.25
6	19	20.00
7	21	22.25
8	23	24.00
9	25	26.00

Pulleys from which considerable power is to be taken should always be placed as close to a bearing as possible.

58. The diameters of the different lengths of shafts composing a line of shafting may be proportional to the quantity of power delivered by each respective length. In this connection it is to be observed that the positions of the various pulleys in reference to the bearings must be taken into consideration in deciding upon the size of a shaft. Suppose, for example, that a piece of shafting delivers a certain amount of power; then, it is obvious that the shaft will deflect or bend less if the pulley transmitting that power be placed close to a hanger or bearing than if it be placed midway between the two hangers or bearings. It is impossible to give any rule for the proper distance of bearings that could be used universally, as in some cases the requirements demand that the bearings be nearer than in others.

Wherever possible it is advisable to have the main line of shafting run through the center of the room, or at least far enough from either wall to allow countershafts to be placed on either side of it. When this is done, power may be taken off the main shaft from either side by alternate pulleys, and the deflection caused in the main shaft in one direction by one pulley will be counteracted by the deflection caused in the opposite direction by the next pulley.

If the work done by a line of shafting is distributed quite equally along its entire length and the power can be applied near the middle, the strength of the shaft need be only half as great as would be required if the power were applied at one end.

59. *To compute the horsepower that can be transmitted by a shaft of any given diameter:*

Let D = diameter of shaft;
R = revolutions per minute;
H = horsepower transmitted;
C = constant given in Table XIV.

TABLE XIV.

CONSTANTS FOR LINE SHAFTING.

Material of Shaft.	No Pulleys Between Bearings.	Pulleys Between Bearings.
Steel or cold-rolled iron.........	65	85
Wrought iron..................	70	95
Cast iron.....................	90	120

In the above table the bearings are supposed to be spaced so as to relieve the shaft of excessive bending; also, in the third vertical column, an average number and weight of pulleys and power given off is assumed.

60. Rules and Formulas.—In determining the above constants, allowance has been made to insure the stiffness as well as strength of the shaft. Cold-rolled iron is considerably stronger than ordinary turned wrought iron; the increased strength is due to the process of rolling, which seems to compress the metal and so make it denser, not merely skin deep, but practically throughout the whole diameter. We have, then, the following

Rule 11.—*The horsepower that a shaft will transmit equals the product of the cube of the diameter and the number of revolutions, divided by the value of C for the given material.*

Or, $$H = \frac{D^3 \times R}{C}.$$

Example.—What horsepower will a 3-inch wrought-iron shaft transmit which makes 100 revolutions per minute, there being no pulleys between bearings?

Solution.—Applying rule **11** and substituting, we have

$$H = \frac{3 \times 3 \times 3 \times 100}{70} = 38.57 \text{ H. P.} \quad \text{Ans.}$$

If there were the usual amount of power taken off, as mentioned above, we should take $C = 95$.

Then, $$H = \frac{27 \times 100}{95} = 28.42 \text{ H. P.} \quad \text{Ans.}$$

61. To compute the number of revolutions a shaft must make to transmit a given horsepower:

Rule 12.—*The number of revolutions necessary for a given horsepower equals the product of the value of C for the given material and the number of horsepower, divided by the cube of the diameter.*

Or, $$R = \frac{C \times H}{D^3}.$$

Example.—How many revolutions must a 3-inch wrought-iron shaft make per minute to transmit 28.42 horsepower, power being taken off at intervals between the bearings?

Solution.—Applying the rule just given and substituting, we have

$$R = \frac{95 \times 28.42}{3 \times 3 \times 3} = 100 \text{ rev.} \quad \text{Ans.}$$

62. To compute the diameter of a shaft that will transmit a given horsepower, the number of revolutions the shaft makes per minute being given:

Rule 13.—*The diameter of a shaft equals the cube root of the quotient obtained by dividing the product of the value of C for the given material and the number of horsepower by the number of revolutions.*

Or, $$D = \sqrt[3]{\frac{C \times H}{R}}.$$

Example.—What must be the diameter of a wrought-iron shaft to transmit 38.57 horsepower, the shaft to make 100 revolutions per minute, no power being taken off between bearings?

Solution.—By rule **13,** we have

$$D = \sqrt[3]{\frac{70 \times 38.57}{100}} = \sqrt[3]{27} = 3 \text{ in.} \quad \text{Ans.}$$

63. As the speed of shafting is used as a multiplier in the calculations of the horsepower of shafts, it is readily seen that a shaft having a given diameter will transmit more power in proportion as its speed is increased. Thus, a shaft that is capable of transmitting 10 horsepower when making 100 revolutions per minute will transmit 20 horsepower when making 200 revolutions per minute. We may, therefore, say *the number of horsepower transmitted by a shaft is directly proportional to the number of revolutions.*

EXAMPLES FOR PRACTICE.

1. What horsepower will a 2½-inch wrought-iron shaft transmit when running at 110 revolutions per minute, it being used for transmission only? Ans. 24.55 H. P.

2. A 6-inch cast-iron shaft transmits 150 horsepower. How many revolutions per minute must it make, no power being taken off between bearings? Ans. 62½ R. P. M.

3. What should be the diameter of a wrought-iron shaft to transmit 100 horsepower at 150 revolutions per minute, power being taken off between bearings? Ans. 4 in., nearly.

4. The machines driven by a certain line of wrought-iron shafting take their power from various points between the bearings, and if all were working together at their full capacity, they would require 65 horsepower to drive them. What diameter should the shaft be if it runs at 150 revolutions per minute? Ans. 3½ in., nearly.

HEAT AND STEAM.

HEAT.

NATURE OF HEAT.

1. All modern scientists and investigators agree that **heat** is a *form of energy*. It is conceived to be a motion of the molecules composing matter. All matter is composed of molecules, which, according to the generally accepted theory, are not in a state of rest, but are moving or vibrating back and forth with a greater or less velocity. It is this movement of the molecules that is generally believed to cause the sensations of warmth and cold; if the motion is slow, the body feels cold; whereas, if the motion is rapid, the body feels warm. Since a body in motion has kinetic energy and since the molecules composing matter are supposed to be in motion, each molecule possesses kinetic energy; hence, we can conceive heat to be a form of energy.

2. **Temperature** is a term used to indicate how hot or cold a body is; i. e., to indicate the velocity of the vibration of the molecules of a body. A body having a high temperature is said to be hot; a body having a low temperature is said to be cold. When a body, as, for example, an iron bar, receives heat from any source, its temperature rises; on the other hand, when a body loses heat, its temperature falls.

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The temperature is *not* a measure of the *quantity* of heat a body possesses. *Temperature* may be considered to be a measure of the velocity with which the molecules of a body vibrate to and fro, while the *quantity of heat* may be considered to be the total energy of the molecules composing the body. A small iron rod may be heated to whiteness and yet possess a very small quantity of heat. Its temperature is very high, but this simply indicates that the molecules of the rod are vibrating with an extremely high velocity. An iron ball 1 foot in diameter and an iron ball 1 inch in diameter may have exactly the same *temperature*, but the larger ball would have by far the greater quantity of heat.

3. The Thermometer.—Temperature is measured by an instrument called the **thermometer,** which is so familiar as to scarcely need description. It consists of a thin glass tube, at one end of which is a bulb filled with mercury. Upon being heated, the mercury expands in proportion to the rise of temperature. Thermometers are graduated in different ways. In the Fahrenheit thermometer, which is the one generally used in this country, the point where the mercury stands when the instrument is placed in melting ice is marked 32°. The point indicated by the mercury when the thermometer is placed in water boiling in the open air at the level of the sea is marked 212°. The tube between these two points is divided into 180 equal parts, called **degrees.**

4. Effects of Heat.—Suppose we take a vessel filled with some substance, say water. Let the vessel be a cylinder fitted with a piston, as shown in Fig. 1. The water is, say, at the freezing point, and the millions of molecules composing the water are moving to and fro with a comparatively small velocity. Place the vessel over a fire or furnace. Heat is communicated to the molecules of water, and they begin to move faster and faster. That is, their kinetic energy increases, and if a thermometer is inserted in the vessel, it will be found that the temperature of the water rises. Consequently, one effect of heat is to raise the temperature of the body to which it is applied. But, after reaching a certain temperature, the molecules of the water not only move faster, but they move farther from each other, and their paths are longer. It is plain that if the molecules are farther apart than they were originally, the whole body of them must take up more space. In other words, after reaching a certain temperature, the water expands as heat is added. Hence, another effect of heat is to cause bodies to expand. Common examples of the expansion of bodies by heat are seen in the setting of tires, the expansion of the rails of a railway in summer, etc.

5. The heat supplied to the vessel of water has so far done three things: (1) It has raised the temperature of the water and thus has increased the kinetic energy of the molecules. Let the amount of heat expended for this purpose be denoted by S. (2) A certain quantity of heat has been used in expanding the water, that is, in pushing the molecules farther apart against the force of cohesion. Denote the amount of heat so expended by I. (3) Since the water expands, it must raise the piston P against the pressure of the atmosphere, and consequently more heat must be used to expand the water than would be required if there were no pressure on the upper side of the piston. Call this extra quantity of heat W.

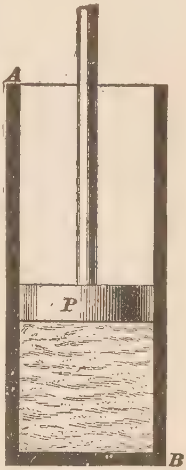

FIG. 1.

If we denote by Q the total quantity of heat given to the vessel of water, we have

$$Q = S + I + W.$$

Ordinarily, the greater part of the heat given to a body is spent in raising its temperature, and but little is used in expanding the body. That is, the quantity S is nearly equal to the quantity Q, while the quantities I and W are extremely small.

6. Suppose that the piston in Fig. 1 is removed from the cylinder, so that the water will be in contact with the atmosphere, and that a thermometer is inserted. As the water becomes more and more heated, the temperature indicated by the thermometer will rise until it reaches 212°. So far, most of the heat has been used to raise the temperature of the water. But now, no matter how much heat is added to the water, the mercury stands at 212°, and cannot be made to rise higher. This is the reason: When the temperature reaches 212° the molecules of water have been set into such rapid motion that the force of cohesion is no longer able to hold them and they tend to separate. In other words, the water changes to a gas (steam), and all the heat is being used to effect this change. The temperature of the steam will remain at 212° until all water is changed to steam; then, if more heat is applied, the temperature of the steam will begin to rise.

7. Again, suppose we take a block of ice at a temperature of, say, 14° and heat it. If a thermometer is placed in contact with the ice, the mercury will rise until it reaches 32° and will then remain stationary. As soon as this temperature is reached, the ice begins to melt or change to water, and the heat, instead of raising the temperature farther, is all used to effect this change of state. Here, then, is another effect produced by heat. It will change a solid to a liquid or a liquid to a gas.

8. Summary.—By a study of the results of applying heat to the substances that have just been considered, we see that the effects of heat most commonly observed are: (1) It increases the rate of motion of the molecules, an effect that is indicated by an increase in temperature. (2) It increases the lengths of the paths of, and the distance between, the molecules, thus causing the body to expand and fill a greater space. (3) It overcomes the attractive forces that tend to hold the molecules of a substance together and thus changes it from a solid to a liquid or from a liquid to a gas, according to the state it was in when the heat was applied.

The second statement of the summary, while generally true, is subject to exceptions, the most notable one of which is water, which in rising from a temperature of 32° Fahrenheit to a temperature of 39.2° contracts instead of expands.

9. Latent Heat.—The heat that is expended in changing a body from the solid to the liquid state or from the liquid to the gaseous state is called **latent heat.** The portion of the heat applied that raises temperature and that, therefore, affects the thermometer is sometimes called **sensible heat.**

10. Measurement of Heat.—Since heat is not a substance, it cannot be measured directly in pounds or quarts, but, like force, it may be measured by the effects it produces. Suppose that a certain quantity of heat raises the temperature of 1 pound of water from 52° to 53° Fahrenheit. It will take the same quantity of heat to raise that pound from 53° to 54°, and therefore it will take nearly double that quantity to raise the temperature of 1 pound of water from 52° to 54°. The unit quantity of heat is the quantity required to raise the temperature of a pound of water from 62° to 63°. This unit is called the **British thermal unit,** and is commonly abbreviated to B. T. U.

11. For temperatures above 63°, it takes slightly more than 1 B. T. U. to produce a change of 1 degree in 1 pound of water—the difference increasing the farther the temperature is from 63°. For temperatures below 62°, it takes slightly less than 1 B. T. U. to produce a change of 1 degree in 1 pound of water; and, as before, the difference is greater the farther the temperature is from 62°. Thus it will take more heat to raise the temperature of 1 pound of water from 75° to 76° than it will to raise it from 74° to 75°. Likewise, it will take less heat to raise the temperature of 1 pound of water from 42° to 43° than to raise it from 43° to

44°. However, the difference between the actual B. T. U., as defined above, and the quantity of heat required to change the temperature of 1 pound of water 1 degree for any other temperature is so small that, for all ordinary purposes, it may be assumed that it takes 1 B. T. U. to produce a change of 1 degree in the temperature of 1 pound of water for all temperatures that we are likely to meet in practice.

12. Relation Between Heat and Work.—Suppose that, in the experiment shown in Fig. 1, the piston had been allowed to remain in the cylinder while the water was being changed to steam. Steam at 212° occupies nearly 1,700 times the space that the water originally occupied. Hence, the piston would be lifted in the cylinder to give room for the steam that was being formed. But to raise the piston requires work. Here, then, is an example of work being performed by heat. On the other hand, work will produce heat. If two blocks of wood are rubbed briskly together, they will become warm, and may even ignite. The work done in overcoming friction causes the journals and bearings of fast-running machines to heat. A small iron rod may be heated to redness by pounding it on an anvil.

13. Since work may be changed into heat and heat into work, it seems probable that there is some fixed ratio between the unit of heat (B. T. U.) and the unit of work, the foot-pound. By a series of careful experiments, Dr. Joule, of England, discovered this ratio.

He found that 1 B. T. U. is equivalent to 772 foot-pounds; later and more careful experiments show that 778 foot-pounds is more nearly correct. This number, 778 foot-pounds, is called the **mechanical equivalent** of 1 B. T. U.

We have, then, the following important law: *Heat may be changed to work or work to heat; 778 foot-pounds of work are required to produce 1 B. T. U.; and, conversely, the expenditure of 1 B. T. U. produces 778 foot-pounds of work.*

EXAMPLE 1.—The burning of 1 pound of coal gives out sufficient heat to raise 14,000 pounds of water from 62° to 63°. If all this heat is utilized, how high will it lift a weight of 700 pounds?

Solution.—Since 1 B. T. U. raises the temperature of 1 pound of water from 62° to 63°, it requires 14,000 B. T. U. to raise 14,000 pounds of water from 62° to 63°; hence, the burning of 1 pound of coal gives out 14,000 B. T. U. One B. T. U. is equivalent to 778 foot-pounds; hence, 14,000 B. T. U. are equivalent to $14{,}000 \times 778 = 10{,}892{,}000$ foot-pounds. Then, the height to which the weight can be raised is $10{,}892{,}000 \div 700 = 15{,}560$ ft. Ans.

Example 2.—A cannon ball weighing 60 pounds moves with a velocity of 1,300 feet per second. Suppose the ball were suddenly stopped and all its kinetic energy changed into heat, how many B. T. U. would be developed? If all this heat were applied to 100 pounds of water at a temperature of 60°, to what temperature would the water be raised?

Solution.—The kinetic energy of the cannon ball is $\frac{W v^2}{64.32} = \frac{60 \times 1{,}300^2}{64.32} = 1{,}576{,}492$ foot-pounds. But 778 foot-pounds = 1 B.T.U. Therefore, the number of B. T. U. developed is $1{,}576{,}492 \div 778 = 2{,}026.3$ B. T. U. Since 1 B. T. U. raises the temperature of 1 pound of water 1 degree, it will take 100 B. T. U. to raise 100 pounds of water 1 degree. Hence, 2,026.3 B. T. U. will raise 100 pounds of water $2{,}026.3 \div 100 = 20.26$ degrees, and the final temperature of the water will be $60° + 20.26° = 80.26°$. Ans.

14. Specific Heat.—One B. T. U. raises the temperature of 1 pound of water 1 degree; will it have the same effect on a pound of mercury? Heat two 1-pound iron balls to the temperature of boiling water, 212°, having now the same weights and temperatures, each ball has the same quantity of heat. Place one of these balls in a vessel, into which slowly pour enough water having a temperature of 60° so that the iron will be cooled to 70° while the water is heated to the same temperature. Now place the other hot ball in another vessel, into which pour mercury having a temperature of 60°, until the iron and mercury reach a common temperature of 70°. In each case the hot ball will have been cooled from 212° to 70°, and therefore each will have given up the same quantity of heat. When, however, we consider the effects produced by the heat, we find that what has been given off by one ball has raised nearly 1.62 pounds of water 10 degrees; that given off by the other ball, which, it will be remembered, is the same amount as in the first case, has raised 48.5 pounds of mercury (or nearly 30 times more mercury than water) the same number of degrees. It is plain, therefore, that to raise 1 pound of mercury from 62° to 63° requires $\frac{1}{30}$ the heat necessary to raise 1 pound of water from 62° to 63°. Hence, we say the specific heat of the mercury is $\frac{1}{30}$ or .0333.

15. *The* **specific heat** *of a body is the ratio between the quantity of heat required to warm that body 1 degree and the quantity of heat required to warm an equal weight of water 1 degree.*

Example 1.—It is found that to raise the temperature of 20 pounds of iron from 62° to 63° requires 2.276 B. T. U. What is the specific heat of iron?

Solution.—To raise 20 pounds of water from 62° to 63° requires 20 B. T. U. The specific heat of the iron is, according to the above definition, the ratio between the quantities of heat required to raise the temperature of the iron and the water, respectively, through 1 degree, that is, it is the ratio $2.276 : 20 = 2.276 \div 20 = .1138$. Ans.

Example 2.—The specific heat of silver is .057. How many B. T. U. are required to raise 22 pounds of silver from 50° to 60°?

Solution.—To raise the temperature of 1 pound of water 1 degree requires 1 B. T. U. Since the specific heat of silver is .057, only .057 B. T. U. is required to raise 1 pound of silver 1 degree. Hence, to raise 22 pounds of silver 10 degrees must require $.057 \times 22 \times 10 = 12.54$ B. T. U. Ans.

Rule 1.—*To find the number of B. T. U. required to raise the temperature of a body a given number of degrees, multiply the specific heat of the body by its weight in pounds and by the number of degrees.*

Denote the number of B. T. U. required by U; the specific heat by c; the weight by W; and let t and t_1, respectively, be the temperatures before and after the heat is applied.

Then, $$U = c\,W(t_1 - t).$$

The specific heat of some of the more common substances is given in the following table:

SPECIFIC HEAT OF SUBSTANCES.

Substance.	Specific Heat.	Substance.	Specific Heat.
Water............	1.0000	Ice..................	.5040
Sulphur...........	.2026	Steam (superheated).	.4805
Iron..............	.1138	Air..................	.2375
Copper............	.0951	Oxygen.............	.2175
Silver.............	.0570	Hydrogen...........	3.4090
Tin................	.0562	Carbon monoxide....	.2479
Mercury...........	.0333	Carbon dioxide......	.2170
Lead..............	.0314	Nitrogen............	.2438

16. Latent Heat of Fusion.—This term is applied to the quantity of heat required to change a pound of a given substance from the solid to the liquid state. The only case of interest to the engineer is the heat required to change 1 pound of ice to water. Careful experiments have shown that about 144 B. T. U. are required to change 1 pound of ice at 32° to water at 32°. Hence, the latent heat of water is 144 B. T. U.

17. The **latent heat of steam** is the quantity of heat required to change 1 pound of water at a given temperature into steam at the same temperature. Experiment has shown that at a temperature of 212° this quantity of heat is about 966 B. T. U. This means that the heat required to change 1 pound of water at 212° to steam is 966 times as great as the heat required to raise the temperature of a pound of water from 62° to 63°. The latent heat of steam is different for different temperatures, as will be seen by referring to the Steam Tables.

EXAMPLE.—How many B. T. U. are required to change 5 pounds of ice at 15° into steam at 212°?

SOLUTION.—To raise the temperature of the ice from 15° to 32° (the melting temperature) requires, according to rule **1**,

$$.504 \times 5 \times (32 - 15) = 42.84 \text{ B. T. U.}$$

To change the ice to water requires 144 B. T. U. for each pound, or 144 × 5 = 720 B. T. U. To raise the water from 32° to 212° requires, from rule **1**,

$$1 \times 5 \times (212 - 32) = 5 \times 180 = 900 \text{ B. T. U.}$$

Finally, to change the water to steam requires 966 B. T. U. per pound, or 966 × 5 = 4,830 B. T. U. Therefore, in all, 42.84 + 720 + 900 + 4,830 = 6,492.84 B. T. U. are required. Ans.

Expressed in foot-pounds, the work required to effect the above change would be 6,492.84 × 778 = 5,051,429.5 foot-pounds, or work enough to lift a weight of 1,000 pounds nearly a mile.

18. Since a pound of ice requires 144 B. T. U. to change it to water, it follows that when a pound of water at 32° changes to ice (freezes), 144 B. T. U. are given out in the process. Similarly, the condensation of a pound of steam into water at 212° liberates 966 B. T. U. This principle is applied in heating buildings by steam. The steam passes through the radiators and condenses. The latent heat thus set free warms the building.

19. Temperatures of Mixtures.—It is often desirable to calculate the final temperature of a mixture of different substances at different temperatures. The following law is to be observed in such cases: *The quantity of heat in a mixture is the same as the quantity of heat contained in the substances before being combined.* If two substances of different temperatures are placed together, they both finally attain the same temperature; the heat lost by one in coming from a higher to a lower temperature is gained by the other in passing from a lower to a higher temperature.

Rule 2.—*To find the temperature of a mixture of several substances, multiply together the weight, specific heat, and temperature of each substance separately and add the products. Next multiply together the weight and specific heat of each of the substances separately and add these products. Divide the former sum by the latter. The result will be the temperature of the mixture.*

EXAMPLE.— 15 pounds of water at 42° and 30 pounds of mercury at 70° are placed in the same vessel, and a ball of lead weighing 19 pounds and having a temperature of 110° is immersed in the mixture. What is the final temperature of the contents?

SOLUTION.—Applying rule **2**, the product of the weight, specific heat, and temperature of the water is $15 \times 1 \times 42 = 630$; of the mercury, $30 \times .0333 \times 70 = 69.93$; of the lead, $19 \times .0314 \times 110 = 65.626$; and the sum is $630 + 69.93 + 65.626 = 765.556$.

The product of the weight and specific heat of the water is $15 \times 1 = 15$; of the mercury, $30 \times .0333 = .999$; of the lead, $19 \times .0314 = .5966$; and the sum is $15 + .999 + .5966 = 16.5956$. Then, the temperature of the mixture is $765.556 \div 16.5956 = 46.13°$. Ans.

20. A particularly important case is the mixture of steam and water. Let W and t_1 represent the weight and temperature of the steam and let w and t represent the weight and temperature of the water. Let T represent the final temperature of the mixture and L the latent heat of the steam at the given temperature. Then, the temperature of a mixture of steam and water may be found by means of the following rule:

Rule 3.—*When steam and water are mixed, the steam condenses. To find the final temperature of the mixture, add the latent heat and the temperature of the steam and multiply this sum by the weight of the steam. To this product add the product of the weight and temperature of the water and divide the sum so obtained by the sum of the weights of the steam and water. The quotient will be the temperature of the mixture.*

Or,
$$T = \frac{W(L + t_1) + w\,t}{W + w}.$$

In the case of a mixture of water and steam, it is to be observed that the temperature of the mixture can never be more than the temperature of the steam, no matter whether the mixture takes place in an open or a closed vessel. The reason for this is to be found in the fact that after the steam has heated the water to its own temperature, there can be no further giving up of heat, since in order that heat may pass from one body to another, there must be a difference in the temperature of the two bodies, the heat passing from the hotter body to the colder one. The only way in which heat can be made to pass from a colder to a hotter body or from one body to another one at an equal temperaature is by the expenditure of work; but as this condition does not ordinarily exist in the case of steam and water being mixed, it follows that the temperature of the mixture can never exceed that of the steam.

Whenever rule **3** gives a temperature in excess of that of the steam, it is an indication that the quantity of water mixed with the steam is not sufficient to condense all the steam, and consequently the given temperature of the steam (in case of a closed vessel) or the temperature of the steam corresponding to the existing atmospheric pressure (in case of an open vessel) should be substituted for the temperature given by rule **3.**

EXAMPLE.—If 8 pounds of steam at 212° are run into a barrel containing 300 pounds of water at 43°, what will be the final temperature?

SOLUTION.—The weight of the steam $W = 8$ pounds; the latent heat $L = 966$ B. T. U.; the temperature $t_1 = 212°$. The weight of the water $w = 300$ pounds; the temperature $t = 43°$. Hence, from rule **3,** the final temperature is

$$\frac{8 \times (966 + 212) + 300 \times 43}{8 + 300} = 72\tfrac{1}{2}°. \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. A body weighing 143 pounds falls 62 feet. If the energy of the body at the end of the fall is changed into heat, how many B. T. U. will be developed? Ans. 11.39 B. T. U.

2. An expenditure of 210 B. T. U. per minute will develop how many horsepower? Ans. 4.95 H. P.

3. If $\frac{1}{7}$ the total heat in the coal is used in doing work, how many pounds of coal must be burned per hour to run a 40-horsepower engine? Each pound of the coal gives out 13,500 B. T. U. Ans. 52.8 lb.

4. A bar of iron weighing 20 pounds and having a temperature of 350° is plunged into a tank containing 130 pounds of water at 55°. To what temperature will the water be raised? Ans. 60°

5. How many pounds of ice at 32° can be melted by 3 pounds of steam at 212°? Ans. 23.875 lb.

Suggestion.—Each pound of ice requires 144 B. T. U. to melt it; each pound of steam in changing to water at 32° gives up 966 + (212 − 32) = 1,146 B. T. U.

6. How many B. T. U. are required to raise the temperature of 26 pounds of copper from 57° to 93°? Ans. 89.01 B. T. U.

7. Four pounds of a certain substance at a temperature of 212° are mixed with 1 pound of water at 50°; the specific heat of the substance being .03125, what is the resulting temperature of the mixture? Ans. 68°.

8. How many B. T. U. are required to change 13 pounds of water at 59° into steam at 212°? Ans. 14,547 B. T. U.

9. Twenty pounds of steam at 212° are run into a tank containing 340 pounds of water at 42°. What will be the final temperature of the mixture? Ans. 105.11°.

STEAM.

PROPERTIES OF STEAM.

21. **Steam** is *water vapor;* that is, it is *water* changed into a *gaseous state* by the application of *heat.*

The process of changing water (or other liquid) into vapor by means of heat is called **evaporation** or **vaporization.**

When an open vessel containing water is placed in contact with fire, the air contained in the water is first driven off and escapes from the surface. The water in contact with the part of the vessel nearest the fire first receives the heat and expands. Its specific gravity is reduced; that is, it becomes lighter than the cooler water above it, and it rises to the surface, cooler water taking its place. In this manner the water keeps up a circulation until it reaches a temperature of 212°. At this stage the molecules nearest the fire attain such a velocity of vibration that they rise through the water above them, overcome the pressure of the air, and escape in the form of a gas. When this occurs the water **boils.**

22. Pressure and Temperature.—It is plain that if the pressure on the surface of the water is increased, it will take more work to force the molecules to the surface; that is, more heat must be given to the water to make it boil, and therefore the boiling point will be raised. We have seen that water exposed to the atmospheric pressure of 14.7 pounds per square inch boils at a temperature of 212°. If the pressure is increased to, say, 32 pounds per square inch, the water will not boil until it reaches a temperature of 254°. On the other hand, if the pressure is lowered to 6 pounds per square inch, the water boils at 170°. Hence, we have the following law:

An increase of pressure on the surface of a liquid raises the boiling point; a decrease of pressure lowers the boiling point.

23. Saturated Steam.—Steam in contact with water is called **saturated steam.** This is the condition of steam in a boiler. Steam at a given pressure is also said to be saturated when its temperature is the same as the temperature at which water boils when subjected to the same pressure; this is true even though the steam is entirely separated from water. According to the law just given, the temperature of saturated steam depends only on the pressure. When the steam in a boiler shows a gauge pressure of 60 pounds, its temperature *must be* 307°. A thermometer placed in a boiler could be used to tell the pressure of the steam. It would be as accurate, though not as convenient, as a steam gauge.

The reader is cautioned against the idea that saturated steam necessarily implies "wet" steam. It may be perfectly free from water particles, but it is saturated if it is in any way in contact with water, so that the pressure and temperature are mutually dependent. In the steam boiler, for example, the space above the water, if viewed through a glass-covered opening, appears to be perfectly transparent, as though filled with air, provided the boiler is not working. This shows that the steam that fills this space is perfectly "dry." When, however, the boiler is steaming rapidly, the violent ebullition may throw up a certain amount of water in the form of a spray, that mingles with the steam, giving it the misty appearance shown in the exhaust from an engine. Steam in this condition is "*wet*" *saturated steam.*

24. In physics the word saturation, whence the term *saturated steam* has been derived, has a meaning somewhat different from that commonly assigned to it. It there means the filling of a space with vapor to that point where condensation begins. Then, we may say that saturated steam is steam subjected to a pressure at which condensation is about to begin; that is, the slightest abstraction of heat or the slightest increase in pressure will cause part of it to condense, and if the steam be separated from the water, the slightest addition of heat will cause it to become superheated.

25. Superheated Steam.—Steam separated from water may be heated like air or any other gas until its temperature is higher than the boiling point corresponding to its pressure.

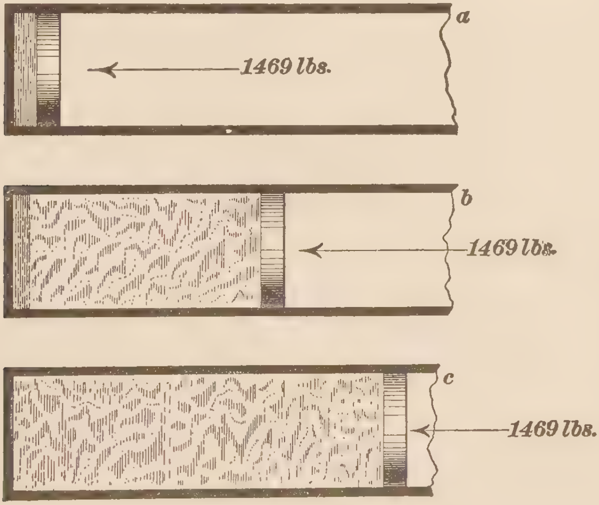

FIG. 2.

To illustrate, put a little water in a cylinder open to the atmosphere, as shown in Fig. 2 at *a*. Suppose that the area of the cylinder is 100 square inches; then, the pressure of the atmosphere upon the piston is 14.69 × 100 = 1,469 pounds. The number 14.69 is a little more exact than 14.7.

When a part of the water is changed to steam, as at *b*, Fig. 2, the steam is in a saturated state, and at this pressure its temperature cannot be higher than 212°. When, however, the water is all changed to steam, as at *c*, Fig. 2, any farther addition of heat raises its temperature, while the pressure remains at 14.69 pounds per square inch. Steam in this latter condition is known as superheated steam.

The specific heat of superheated steam at constant pressure is .4805, or, say, .48 for ordinary purposes; that is, .48 of 1 B. T. U. will raise the temperature of 1 pound of superheated steam 1 degree. The temperature of saturated steam cannot be raised if the pressure remains constant. All the heat is expended in changing water to steam, and until all the water is vaporized the temperature remains constant.

26. If the steam in the cylinder has been subjected to a uniform pressure greater than that of the atmosphere, its temperature will be correspondingly higher; but as long as any water remains, any addition of heat will merely change more of the water to steam without increasing the temperature. As soon, however, as the last drop of water is gone, the effect of adding more heat will be to increase the temperature. We may, therefore, define **superheated steam** as steam separated from water and heated so as to give it a temperature higher than the boiling point corresponding to its pressure.

STEAM TABLES.

PROPERTIES OF SATURATED STEAM.

27. Whenever the pressure of saturated steam is changed, there are other properties that change with it. These properties are the following:

1. The temperature of the steam or, what is the same thing, the boiling point.

2. The number of B. T. U. required to raise 1 pound of water from 32° (freezing) to the boiling point corresponding to the given pressure. This is called the **heat of the liquid.**

3. The number of B. T. U. required to change 1 pound of water at the boiling temperature into steam at the same temperature. This is called the **latent heat of vaporization,** or simply **latent heat.**

4. The number of heat units required to change 1 pound of water at 32° to steam of the required temperature and pressure. This is called the **total heat of vaporization,** or simply **total heat.**

It is plain that the total heat is the sum of the heat of the liquid and the latent heat. That is, total heat = heat of liquid + latent heat.

5. The **specific volume** of the steam at the given pressure; that is, the number of cubic feet occupied by 1 pound of steam of the given pressure.

6. The **density** of the steam; that is, the weight of 1 cubic foot of the steam at the given pressure.

28. All the above properties vary with the pressure of the steam. For example, if steam is at atmospheric pressure, the temperature is at 212°; the heat of the liquid is 180.531 B. T. U.; the latent heat, 966.069 B. T. U.; the total heat, 1,146.6 B. T. U. A pound of steam at this pressure occupies 26.37 cubic feet and 1 cubic foot of the steam weighs about .037928 pound. When the pressure is 70 pounds per square inch above vacuum, the temperature is 302.774°; the heat of the liquid is 272.657 B. T. U.; the latent heat is 901.629 B. T. U.; the total heat, 1,174.286 B. T. U. A pound of the steam occupies 6.076 cubic feet, and 1 cubic foot of the steam weighs .164584 pound.

These properties have been determined by direct experiment for all ordinary steam pressures. They are given in the Table of the Properties of Saturated Steam.

29. Explanation of Table.—Column 1 gives the pressures from 1 to 300 pounds. These pressures are above vacuum. The steam gauges fitted on steam boilers register the pressure above the atmosphere. That is, if the steam is at atmospheric pressure, 14.7 pounds per square inch, the gauge registers 0. Consequently, the atmospheric pressure must be added to the reading of the gauge to obtain the pressure above vacuum. In using the table, care must be taken *not* to use the gauge pressures without first adding 14.7 pounds per square inch for places at sea level. As the pressure of the atmosphere varies with the altitude, the true atmospheric pressure can always be obtained by consulting the barometer. The barometric reading should be reduced to pounds pressure per square inch and added to the gauge pressure in order to obtain the correct absolute pressure for the time and place. However, in nearly all engineering calculations, it is customary to take the pressure of the atmosphere as 14.7 pounds per square inch, which value will hereafter be used unless stated otherwise.

Pressures registered above vacuum are called **absolute pressures.** The pressures given in column 1 are *absolute*.

Column 2 gives the temperature of the steam when at the pressure shown in column 1.

Column 3 gives the **heat of the liquid.** It will be noticed that the values in column 3 may be obtained, approximately, by subtracting 32° from the temperature in column 2. If the specific heat of water were exactly 1.00, it would, of course, take exactly $212 - 32 = 180$ B. T. U. to raise 1 pound of water from 32° to 212°. But experiment shows that the specific heat of water is slightly greater than 1.00 when the temperature of the water is above 62°, and it therefore takes 180.531 B. T. U. to raise 1 pound of water from 32° to 212°.

Column 4 gives the **latent heat of vaporization,** which is seen to decrease slightly as the pressure increases.

Column 5 gives the **total heat of vaporization.** The values in column 5 may be obtained by adding together the corresponding values in columns 3 and 4.

Column 6 gives the weight of 1 cubic foot of steam in pounds. As would be expected, as the pressure increases, the steam becomes denser and weighs more per cubic foot.

Column 7 gives the number of cubic feet occupied by 1 pound of steam at the given pressure. It will be noticed that the corresponding values of columns 6 and 7 multiplied together always produce 1. That is, for 31.3 pounds pressure gauge, $.11088 \times 9.018 = 1.000$, nearly.

Column 8 gives the ratio of the volume of 1 pound of steam at the given pressure and the volume of 1 pound of water at 39.1°. The values in column 8 may be obtained by dividing 62.425, the weight of 1 cubic foot of water at 39.1°, by the numbers in column 6.

30. Directions for Using Steam Table.—Manifestly it would be impossible to compile Steam Tables that would include the values corresponding to all pressures. The table here given covers the range of pressures likely to be met in practice. For values that are not given in the table, though within its range, a method of calculation, known as **interpolation,** may be used. In finding the values of t, q, L, etc. for an absolute pressure of, say, 76.35 pounds per square inch, we must make use of the two values of t, q, L, etc., and also of the two values of p given in the table that are nearest to 76.35 pounds; that is, the nearest given value of p that is less than 76.35 pounds and that which is greater than 76.35 pounds. In the present case, these two values are 76 pounds and 78 pounds. In like manner, for any other given value of p that, though within the range of the table, is not given, we must make use of the two values of p between which it is included.

31. The method of using the tables is illustrated by the following examples and their accompanying solutions.

What are the values of t, q, L, H, V, and W for steam whose gauge pressure is 61.65 pounds per square inch?

The pressures in the Steam Table are absolute pressures; hence, when the gauge pressure of steam is given, 14.7 must be added to it in order to use it in connection with the table.

Therefore,

$$p = 61.65 + 14.7 = 76.35 \text{ pounds per square inch.}$$

Turning to the tables, we find that this pressure lies between the two given values 76 and 78.

(*a*) To find t (temperature):

$$\begin{array}{lll} \text{For } p = & 78 \text{ pounds,} & t = 310.123^\circ \\ \text{For } p = & \underline{76} \text{ pounds,} & t = \underline{308.344^\circ} \\ \text{Difference,} & 2 \text{ pounds,} & \quad\;\; 1.779^\circ \end{array}$$

For a difference in pressure of 2 pounds, we have a difference in temperature of 1.779°. A difference in pressure of 1 pound would, therefore, give a difference in temperature of $\frac{1.779}{2} = .8895^\circ$. The actual difference in pressure for this case is $76.35 - 76 = .35$ lb. Hence, the actual difference in temperature would be $.35 \times .8895 = .311325^\circ$, say, .311°. This means that if the pressure of saturated steam is changed from 76 pounds per square inch to 76.35 pounds per square inch, its temperature is raised through .311°. Its temperature at 76 pounds pressure is 308.344°, consequently, its temperature at 76.35 pounds pressure is $308.344 + .311 = 308.655^\circ$. Ans.

(*b*) To find q (sensible heat):

$$\begin{array}{lll} \text{For } p = & 78 \text{ pounds,} & q = 280.170 \text{ B. T. U.} \\ \text{For } p = & \underline{76} \text{ pounds,} & q = \underline{278.350} \text{ B. T. U.} \\ \text{Difference,} & 2 \text{ pounds,} & \quad\;\; 1.820 \text{ B. T. U.} \end{array}$$

For a difference in 1 pound pressure there will be a difference in q of $\frac{1.820}{2} = .91$ B. T. U.; for a difference in .35 pound pressure there is a difference in q of $.91 \times .35 = .3185$ B. T. U. Hence, the sensible heat q, corresponding to a pressure of 76.35 pounds per square inch, is $278.350 + .3185 = 278.6685$ B. T. U. Ans.

(*c*) To find L (latent heat):

For $p = 76$ pounds, $L = 897.635$ B. T. U.
For $p = \underline{78}$ pounds, $L = \underline{896.359}$ B. T. U.
Difference, 2 pounds, 1.276 B. T. U.

For a difference of 1 pound pressure, the difference in latent heat is $\frac{1.276}{2} = .638$ B. T. U.

$$.638 \times .35 = .2233 \text{ B. T. U.}$$

By comparing p and L in the table, it will be seen that as p increases L decreases. In the present case, therefore, we must subtract from the value of L corresponding to a pressure of 76 pounds the difference due to the actual difference in pressures.

That is,

For $p = 76.35$ pounds, $L = 897.635 - .2233$
$= 897.4117$ B. T. U. Ans.

(*d*) To find H (total heat):

For $p = 78$ pounds, $H = 1{,}176.529$ B. T. U.
For $p = \underline{76}$ pounds, $H = \underline{1{,}175.985}$ B. T. U.
Difference, 2 pounds, .544 B. T. U.

As above, $\frac{.544}{2} = .272$; $.272 \times .35 = .0952$ B. T. U.

Hence,

For $p = 76.35$ pounds, $H = 1{,}175.985 + .0952$
$= 1{,}176.0802$ B. T. U. Ans.

(*e*) To find W (weight per cubic foot):

For $p = 78$ pounds (pressure), $W = .182229$ pound (weight).
For $p = \underline{76}$ pounds (pressure), $W = \underline{.177825}$ pound (weight).
Difference, 2 pounds (pressure), .004404 pound (weight).

Proceeding as before,

$$\frac{.004404}{2} = .002202; \quad .002202 \times .35 = .0007707.$$

Hence,

$$\text{For } p = 76.35 \text{ pounds, } W = .177825 + .0007707$$
$$= .1785957 \text{ pound. Ans.}$$

(*f*) To find V (volume of 1 pound):

$$\begin{array}{lll} \text{For } p = \underline{76} \text{ pounds,} & V = 5.624 \text{ cubic feet.} \\ \text{For } p = \underline{78} \text{ pounds,} & V = \underline{5.488} \text{ cubic feet.} \\ \text{Difference, } 2 \text{ pounds,} & .136 \text{ cubic foot.} \end{array}$$

As before,

$$\frac{.136}{2} = .068; \; .068 \times .35 = .0238 \text{ cubic foot.}$$

Here V decreases as p increases. Hence, as in (*c*), the difference must be subtracted from the value of V corresponding to 76 pounds, and we have

$$\text{For } p = 76.35 \text{ pounds, } V = 5.624 - .0238$$
$$= 5.6002 \text{ cubic feet. Ans.}$$

EXAMPLE 1.—Calculate the heat required to change 5 pounds of water at 32° into steam at 92 pounds pressure above vacuum.

SOLUTION.—From column 5, the total heat of 1 pound at 92 pounds pressure is 1,180.045 B. T. U.

$$1{,}180.045 \times 5 = 5{,}900.225 \text{ B. T. U. Ans.}$$

EXAMPLE 2.—How many heat units are required to raise 8½ pounds of water from 32° to 250° F. ?

SOLUTION.—Looking in column 3, the heat of the liquid of 1 pound at 250.293° is 219.261 B. T. U. 219.261 − .293 = 218.968 B. T. U. = heat of liquid for 250°. Then, for 8½ pounds it is 218.968 × 8½ = 1,861.228 B. T. U. Ans.

EXAMPLE 3.—How many foot-pounds of work will it require to change 60 pounds of boiling water at 80 pounds pressure, absolute, into steam of the same pressure?

SOLUTION.—Looking under column 4, the latent heat of vaporization is 895.108; that is, it takes 895.108 B. T. U. to change 1 pound of water at 80 pounds pressure into steam of the same pressure. Therefore, it takes 895.108 × 60 = 53,706.48 B. T. U. to perform the same operation upon 60 pounds of water.

$$53{,}706.48 \times 778 = 41{,}783{,}641.44 \text{ ft.-lb. Ans.}$$

Example 4.—Find the volume occupied by 14 pounds of steam at 30 pounds *gauge* pressure.

Solution.— 30 pounds gauge pressure = 30 + 14.7 = 44.7 absolute pressure. The nearest pressure in the table is 44 pounds, and the volume of a pound of steam at that pressure is 9.403 cubic feet. The volume of a pound at 46 pounds pressure is 9.018 cubic feet. 9.403 − 9.018 = .385 cubic foot, the difference in volume for a difference in pressure of 2 pounds. $\frac{.385}{2} = .1925$ cubic foot, the difference in volume for a difference in pressure of 1 pound. .1925 × .7 = .135 cubic foot, the difference in volume for a difference in pressure of .7 pound. Therefore, 9.403 − .135 = 9.268 cubic feet is the volume of 1 pound of steam at 44.7 pounds pressure. The .135 cubic foot is subtracted from 9.403 cubic feet, since the volume is less for a pressure of 44.7 pounds than for 44 pounds.

$$9.268 \times 14 = 129.752 \text{ cu. ft.} \quad \text{Ans.}$$

Example 5.—Find the weight of 40 cubic feet of steam at a temperature of 254° F.

Solution.—The weight of 1 cubic foot of steam at 254.002°, from the table, is .078839 pound. Neglecting the .002°, the weight of 40 cubic feet is, therefore,

$$.078839 \times 40 = 3.15356 \text{ lb.} \quad \text{Ans.}$$

Example 6.—How many pounds of steam at 64 pounds pressure, absolute, are required to raise the temperature of 300 pounds of water from 40° to 130° F., the water and steam being mixed together?

Solution.—The number of heat units required to raise 1 pound from 40° to 130° is 130 − 40 = 90 B. T. U. (Actually a little more than 90 would be required, but the above is near enough for all practical purposes.) Then, to raise 300 pounds from 40° to 130° requires 90 × 300 = 27,000 B. T. U. This quantity of heat must necessarily come from the steam. Now, 1 pound of steam at 64 pounds pressure gives up, in condensing, its latent heat of vaporization, or 905.9 B. T. U. But in addition to its latent heat, each pound of steam on condensing must give up an additional amount of heat in falling to 130°. Since the original temperature of the steam was 296.805° F. (see table), each pound gives up by its fall of temperature 296.805 − 130 = 166.805 B. T. U. Therefore, each pound of the steam gives up a total of

$$905.9 + 166.805 = 1{,}072.705 \text{ B. T. U.}$$

It will, therefore, take $\frac{27{,}000}{1{,}072.705} = 25.17$ lb. of steam to accomplish the desired result. Ans.

EXAMPLES FOR PRACTICE.

1. How many foot-pounds of work are required to change 42 pounds of water at the temperature corresponding to a pressure of 88 pounds, absolute, into steam at a temperature corresponding to a pressure of 105 pounds, absolute? Ans. 29,208,194.15 lb.

2. How many B. T. U. are required to convert 25 pounds of water at 32° into 109.6 cubic feet of steam? Ans. 29,541.1 B. T. U.

3. Find the number of heat units required to change 11 pounds of water at 32° into steam at 100 pounds absolute pressure. Ans. 13,000.526 B. T. U.

4. Find the weight of 712 cubic feet of steam at a pressure of 33 pounds, gauge. Ans. 81.689 lb.

5. How many pounds of steam at 47.3 pounds pressure, gauge, are required to raise 120 pounds of water from 55° to 160° at atmospheric pressure? Ans. 12.091 lb.

6. Find the volume of 19 pounds of steam at a pressure of 62 pounds, gauge. Ans. 105.952 cu. ft.

THE PROPERTIES OF SATURATED STEAM.

Pressure above Vacuum in Pounds per Square Inch.	Temperature, Fahrenheit Degrees.	Quantities of Heat in British Thermal Units.			Weight of a Cubic Foot of Steam in Pounds.	Volume.	
		Required to Raise Temperature of the Water from 32° to $t°$.	Total Latent Heat at Pressure P.	Total Heat above 32°.		Of a Pound of Steam in Cubic Feet.	Ratio of Vol. of Steam to Vol. of Eq. Weight of Dist. Water at Temp. of Maximum Density
1	2	3	4	5	6	7	8
p	t	q	L	H	W	V	R
1	102.018	70.040	1043.015	1113.055	.003027	330.4	20623
2	126.302	94.368	1026.094	1120.462	.005818	171.9	10730
3	141.654	109.764	1015.380	1125.144	.008522	117.3	7325
4	153.122	121.271	1007.370	1128.641	.011172	89.51	5588
5	162.370	130.563	1000.899	1131.462	.013781	72.56	4530
6	170.173	138.401	995.441	1133.842	.016357	61.14	3816
7	176.945	145.213	990.695	1135.908	.018908	52.89	3302
8	182.952	151.255	986.485	1137.740	.021436	46.65	2912
9	188.357	156.699	982.690	1139.389	.023944	41.77	2607
10	193.284	161.660	979.232	1140.892	.026437	37.83	2361
11	197.814	166.225	976.050	1142.275	.028911	34.59	2159
12	202.012	170.457	973.098	1143.555	.031376	31.87	1990
13	205.929	174.402	970.346	1144.748	.033828	29.56	1845
14	209.604	178.112	967.757	1145.869	.036265	27.58	1721
14.69	212.000	180.531	966.069	1146.600	.037928	26.37	1646
15	213.067	181.608	965.318	1146.926	.038688	25.85	1614
16	216.347	184.919	963.007	1147.926	.041109	24.33	1519
17	219.452	188.056	960.818	1148.874	.043519	22.98	1434
18	222.424	191.058	958.721	1149.779	.045920	21.78	1359
19	225.255	193.918	956.725	1150.643	.048312	20.70	1292

1	2	3	4	5	6	7	8
p	*t*	*q*	*L*	*H*	*W*	*V*	*R*
20	227.964	196.655	954.814	1151.469	.050696	19.73	1231.0
22	233.069	201.817	951.209	1153.026	.055446	18.04	1126.0
24	237.803	206.610	947.861	1154.471	.060171	16.62	1038.0
26	242.225	211.089	944.730	1155.819	.064870	15.42	962.3
28	246.376	215.293	941.791	1157.084	.069545	14.38	897.6
30	250.293	219.261	939.019	1158.280	.074201	13.48	841.3
32	254.002	223.021	936.389	1159.410	.078839	12.68	791.8
34	257.523	226.594	933.891	1160.485	.083461	11.98	748.0
36	260.883	230.001	931.508	1161.509	.088067	11.36	708.8
38	264.093	233.261	929.227	1162.488	.092657	10.79	673.7
40	267.168	236.386	927.040	1163.426	.097231	10.28	642.0
42	270.122	239.389	924.940	1164.329	.101794	9.826	613.3
44	272.965	242.275	922.919	1165.194	.106345	9.403	587.0
46	275.704	245.061	920.968	1166.029	.110884	9.018	563.0
48	278.348	247.752	919.084	1166.836	.115411	8.665	540.9
50	280.904	250.355	917.260	1167.615	.119927	8.338	520.5
52	283.381	252.875	915.494	1168.369	.124433	8.037	501.7
54	285.781	255.321	913.781	1169.102	.128928	7.756	484.2
56	288.111	257.695	912.118	1169.813	.133414	7.496	467.9
58	290.374	260.002	910.501	1170.503	.137892	7.252	452.7
60	292.575	262.248	908.928	1171.176	.142362	7.024	438.5
62	294.717	264.433	907.396	1171.829	.146824	6.811	425.2
64	296.805	266.566	905.900	1172.466	.151277	6.610	412.6
66	298.842	268.644	904.443	1173.087	.155721	6.422	400.8
68	300.831	270.674	903.020	1173.694	.160157	6.244	389.8
70	302.774	272.657	901.629	1174.286	.164584	6.076	379.3
72	304.669	274.597	900.269	1174.866	.169003	5.917	369.4
74	306.526	276.493	898.938	1175.431	.173417	5.767	360.0
76	308.344	278.350	897.635	1175.985	.177825	5.624	351.1
78	310.123	280.170	896.359	1176.529	.182229	5.488	342.6
80	311.866	281.952	895.108	1177.060	.186627	5.358	334.5
82	313.576	283.701	893.879	1177.580	.191017	5.235	326.8
84	315.250	285.414	892.677	1178.091	.195401	5.118	319.5
86	316.893	287.096	891.496	1178.592	.199781	5.006	312.5
88	318.510	288.750	890.335	1179.085	.204155	4.898	305.8

1	2	3	4	5	6	7	8
p	t	q	L	H	W	V	R
90	320.094	290.373	889.196	1179.569	.208525	4.796	299.4
92	321.653	291.970	888.075	1180.045	.212892	4.697	293.2
94	323.183	293.539	886.972	1180.511	.217253	4.603	287.3
96	324.688	295.083	885.887	1180.970	.221604	4.513	281.7
98	326.169	296.601	884.821	1181.422	.225950	4.426	276.3
100	327.625	298.093	883.773	1181.866	.230293	4.342	271.1
105	331.169	301.731	881.214	1182.945	.241139	4.147	258.9
110	334.582	305.242	878.744	1183.986	.251947	3.969	247.8
115	337.874	308.621	876.371	1184.992	.262732	3.806	237.6
120	341.058	311.885	874.076	1185.961	.273500	3.656	228.3
125	344.136	315.051	871.848	1186.899	.284243	3.518	219.6
130	347.121	318.121	869.688	1187.809	.294961	3.390	211.6
135	350.015	321.105	867.590	1188.695	.305659	3.272	204.2
140	352.827	324.003	865.552	1189.555	.316338	3.161	197.3
145	355.562	326.823	863.567	1190.390	.326998	3.058	190.9
150	358.223	329.566	861.634	1191.200	.337643	2.962	184.9
160	363.346	334.850	857.912	1192.762	.358886	2.786	173.9
170	368.226	339.892	854.359	1194.251	.380071	2.631	164.3
180	372.886	344.708	850.963	1195.671	.401201	2.493	155.6
190	377.352	349.329	847.703	1197.032	.422280	2.368	147.8
200	381.636	353.766	844.573	1198.339	.443310	2.256	140.8
210	385.759	358.041	841.556	1199.597	.464295	2.154	134.5
220	389.736	362.168	838.642	1200.810	.485237	2.061	128.7
230	393.575	366.152	835.828	1201.980	.506139	1.976	123.3
240	397.285	370.008	833.103	1203.111	.527003	1.898	118.5
250	400.883	373.750	830.459	1204.209	.547831	1.825	114.0
260	404.370	377.377	827.896	1205.273	.568626	1.759	109.8
270	407.755	380.905	825.401	1206.306	.589390	1.697	105.9
280	411.048	384.337	822.973	1207.310	.610124	1.639	102.3
290	414.250	387.677	820.609	1208.286	.630829	1.585	99.0
300	417.371	390.933	818.305	1209.238	.651506	1.535	95.8

THE STEAM ENGINE.

INTRODUCTION.

CLASSIFICATION OF STEAM ENGINES.

1. The great number of the types of the steam engine may be classified as follows:

1. According to the kind of service, as *stationary*, *locomotive*, *marine*, *etc.*

2. According to number and arrangement of cylinders, as *simple*, *compound*, *triple expansion*, *quadruple expansion*, *duplex*, *etc.*

3. According to the type of valve used to control the distribution of steam, as *plain slide valve*, *automatic cut-off*, *Corliss*, *etc.*

4. According to the motion of the piston, as *reciprocating*, *rotary*.

Each of these types may be horizontal or vertical, condensing or non-condensing, and, except the rotary engine, single-acting or double-acting.

2. All the different types of reciprocating engines involve essentially the same principles, and therefore the description of a single type will be sufficient to give a general knowledge of these principles. For this purpose we shall choose the simple slide-valve engine, which is the engine in most common use.

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THE RECIPROCATING STEAM ENGINE.

3. Steam may do work by acting on a piston working in a cylinder so as to lift weights or overcome the pressure of the atmosphere; in most cases, however, the work can best be done through the action of shafts and wheels having a continuous rotary motion; it is therefore essential that some method of changing the to-and-fro, or reciprocating, motion of the piston into a continuous rotary motion should be devised. The form of mechanism used for this purpose in practically all types of reciprocating engines is shown in diagrammatic form in Fig. 1.

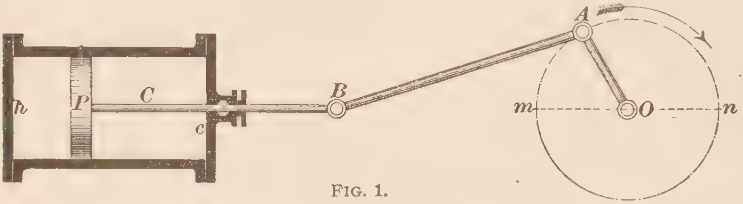

FIG. 1.

The steam from the boiler enters one end—say, in this case, the end *h*—of the **cylinder,** and pushes the piston to the other end. By means of another mechanism, called the **valve,** the steam is now admitted to the end *c* of the cylinder, while the end *h* is at the same time allowed to communicate with the atmosphere or with a condenser. The steam in *h* escapes, while that in *c* pushes back the piston to its original position, whence the same operation is repeated.

Attached to the piston, and forming a part of it, is the piston rod *C B*; to the end of *C B* is fastened, by a joint, one end of the link *B A*. The other end of *B A* is joined to the link *A O*; and the other end of *A O* terminates in a shaft *O*, which is fixed in stationary bearings. It is evident that the end of *B A* which is attached to *C B* can move only in a straight line; and since the shaft *O* can only rotate in its bearings, the end of *A O* which is attached to *B A* can move only in a circle.

4. When the piston is at one extreme end of the cylinder, say at *h*, the joint *A* is at the point *m*, and all three links, $A\,O$, $B\,A$, and $C\,B$, lie in a straight line. As the piston moves to the right, the link $C\,B$ moves also to the right, while the joint A must move in a semicircle $m\,n$. When P arrives at the other end of the cylinder, the joint A is at n, and again $A\,O$, $B\,A$, and $C\,B$ are in a straight line. The piston now moves back to the end h of the cylinder, the joint A moving in the other semicircle from n to m.

The link $A\,O$ is called the **crank,** $B\,A$ the **connecting-rod,** and $C\,B$ the **piston rod.** Those parts that have a to-and-fro, or reciprocating, motion are called the **reciprocating parts.**

The end h of the cylinder is called the **head end,** and the end c the **crank end.** The distance passed over by the piston during half a revolution of the crank is called the **stroke,** and is plainly equal to the diameter of the circle described by the end of the crank—that is, the distance $m\,n$.

5. The engine may run in the direction shown by the arrow in the figure or it may run in the reverse direction. In the former case it is said to *run over* and in the latter case to *run under.*

The stroke from the head end to the crank end of the cylinder, that is, from left to right in the figure, is called the **forward stroke;** the one from crank end to head end, the **return stroke.**

The above simple mechanism perfectly fulfils the office of giving a continuous rotary motion in one direction. A pulley is keyed to the shaft O and the power is transferred by belting from the pulley to shafting or directly to the machinery to be run.

CONSTRUCTION OF A PLAIN SLIDE-VALVE ENGINE.

6. In Fig. 2, the construction of a **plain slide-valve engine** is shown, and in Fig. 3 is shown a section of a steam cylinder. Referring to these figures, H is the head end and C the crank end of the steam cylinder; B and B' are the steam ports; D is the steam chest; E is the exhaust

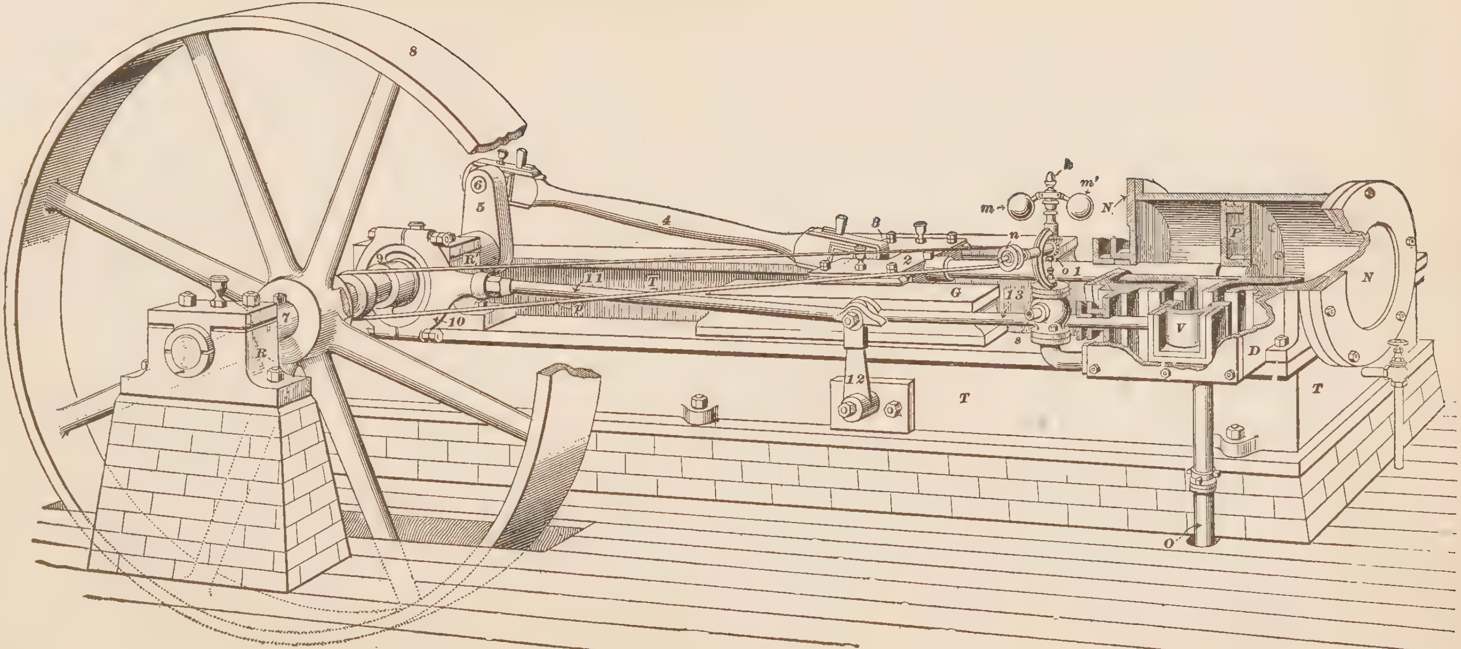

FIG. 2.

port; *N* and *N'* are the cylinder heads; *S* is the steam-supply pipe; *O* is the exhaust pipe, and connects with the exhaust port *E*; *G* is one of the two guide bars; *R* and *R'* are the shaft bearings; and *T* is the bed, or frame, of the engine. The above are all **stationary parts** of the engine, or parts that do not change their relative positions when the engine is in motion. *P* is the piston; *1* is the piston rod; *2* is the crosshead; *3* is the crosshead pin, which is often called the wristpin; *4* is the connecting-rod; *5* is the crank; *6* is the crankpin; *7* is the crank-shaft; *8* is the flywheel; *9* is the eccentric; *10* is the eccentric strap; *11* is the eccentric rod; *12* is the rocker; *13* is the valve rod or stem; and *V* is the slide valve. These are all movable parts of the engine, or parts that change their relative positions when the engine is in motion. The eccentric, eccentric strap, eccentric rod, rocker, valve stem, and slide valve, which form the mechanism by means of which the steam is distributed, when considered as a whole, are termed the **valve gear** of the engine.

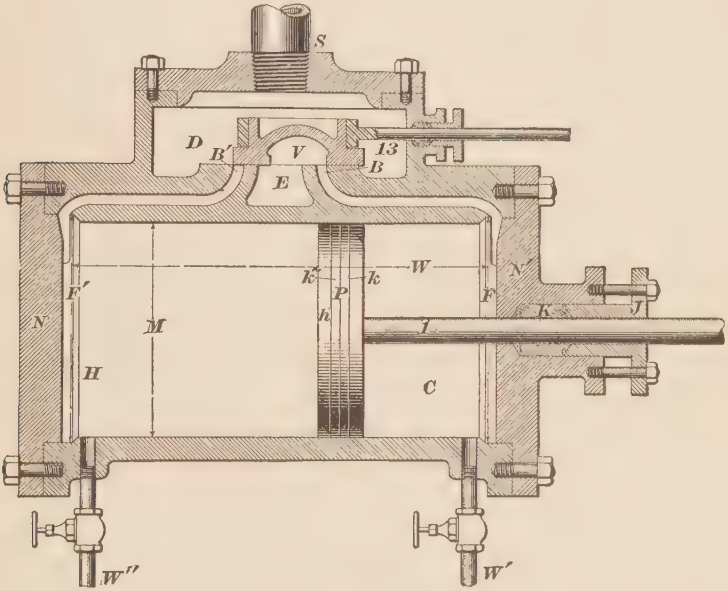

FIG. 3.

7. The working length of the cylinder is shown by the dimension line *W*. It is slightly less than the distance between the cylinder heads, since a small space must be left between the head and the piston, when the latter is at the end of its stroke. The stroke of the engine is the travel of the piston *P*; since the piston and crosshead are rigidly fastened to the same rod, the stroke must also be equal to the travel of the crosshead. It was shown in Fig. 1 that the stroke is also equal to the diameter of the circle described by the crankpin *6*, or, what is the same thing, it is equal to twice the length of the crank *5*, this length being measured from the center of the crankpin *6* to the center of the crankshaft *7*. The diameter or bore of the cylinder is represented by *M*.

8. The **size** of an engine is generally expressed by giving the diameter of the cylinder and the stroke in inches. Thus, an engine having a cylinder diameter of 16 inches and a stroke of 22 inches is called a 16″ × 22″ engine.

9. At the ends *F* and *F′* the cylinder is *counterbored*—that is, for a short distance the bore is greater than *M*. The piston projects partly into this counterbore at the end of each stroke. Were it not for the counterbore, the piston would not wear the cylinder walls their entire length, and shoulders would be formed at each end of the cylinder. When the wear of the joints in the connecting-rod is taken up, the length of the connecting-rod is slightly increased, and the piston is shoved back slightly towards the head end of the cylinder. In this case, the shoulder would cause an undesirable pounding of the piston.

Drain cocks *W′* and *W″*, Fig. 3, are fitted to each end of the cylinder, through which any condensed steam may be discharged.

10. The piston fits loosely in the cylinder and has split rings *k* and *k′* inserted, which spring out so as to press against the walls of the cylinder and prevent leakage of steam between the wall of the cylinder and piston. Pistons are usually supplied with a follower plate *h*, which is bolted to the head end of the piston *P* in order to hold these split rings *k* and *k'* in place. The piston rod *1* is a round bar rigidly connected to both the piston *P* and the crosshead *2*.

11. A stuffingbox *K* in which packing is placed is fitted with a gland *J*, which, when bolted down, compresses the packing around the piston rod *1* and makes a steam-tight joint. This packing is usually made in the form of split rings, which are so placed that the split of the first ring is covered by the solid part of the next ring. When repacking, care should be taken not to cause unnecessary friction by too much pressure from the gland. The guide bars, as *G*, Fig. 2, constrain the crosshead *2* to move exactly in line with the axis of the cylinder, thus relieving the piston rod of all bending stresses.

The connecting-rod *4* forms the connecting link between the crosshead and crank *5*. The joint between crosshead *2* and connecting-rod *4* is made by the crosshead pin *3*, and that between the connecting-rod and crank by the crank-pin *6*. Connecting-rods are usually made from 4 to 6 times the length of the crank, or from 4 to 6 "cranks" in length.

THE ECCENTRIC.

12. Fig. 4 shows the eccentric that imparts motion to the slide valve *V* in Figs. 2 and 3. It consists of a circular disk of iron *a*, which is keyed or fastened by setscrews to the shaft and revolves with it. The center of this disk, which is called the **eccentric sheave,** is at *O*. It is evident that, as the shaft revolves, the center *O* of the sheave *a* will describe the dotted circle *b*, whose center is the center of the shaft. Consequently, the eccentric strap *c* and the eccentric rod *d*, to which it is fastened, will be moved horizontally, during a half revolution, a distance equal to the diameter *e* of the dotted circle. This distance *e* is commonly called the **throw** of the eccentric. The distance *O Q* between the center of the eccentric and center of the shaft is called the **radius** of the eccentric or the **eccentricity**. It is plain that the throw is *twice* the radius. Attention is here called to the fact that practice varies somewhat in the definition of the term *throw*. Some engineers call the radius the throw, but by far the greater part define throw as here given.

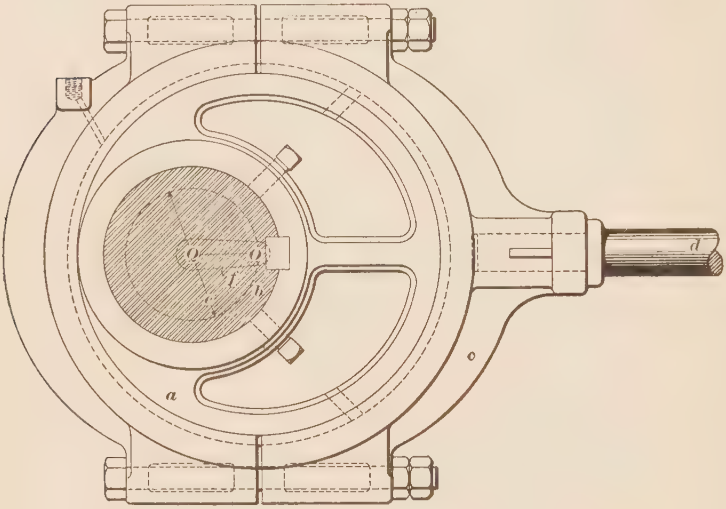

FIG. 4.

13. The eccentric is equivalent to a crank whose length is equal to the radius of the eccentric. Thus, if the end of the eccentric rod *d* were attached at *O* to the crank *f* (shown in dotted lines), the crank would give the same motion to the rod that the eccentric does. In plain slide-valve engines, the eccentric is usually keyed to the shaft after being properly adjusted.

The connection between the eccentric rod *11*, Fig. 2, and the valve stem *13* is accomplished in a variety of ways. In Fig. 2, a rocker-arm *12* is used to support the joint between the eccentric rod *11* and the valve stem *13*. The latter must be supported in some manner to prevent its binding in its stuffingbox.

14. Motion of Eccentric and Valve.—As the motion of the valve is given by the eccentric, the valve is in mid-position in a horizontal engine when the radius of the eccentric is in a vertical position. When *Q O*, Fig. 4, lies horizontally on the right side of *Q*, the valve *V* (see Fig. 2) is in its position nearest the head end of the steam chest, and when *O Q* lies horizontally on the left side of *Q*, the valve is at the end of its stroke towards the crank end of the steam chest.

THE D SLIDE VALVE AND STEAM DISTRIBUTION.

ACTION OF SLIDE VALVE.

15. Description of the Slide Valve.—Of the different kind of valves used to distribute the steam in the engine cylinder, the **D** slide valve is the most common. A section of such a valve is shown in Fig. 5 in its central position;

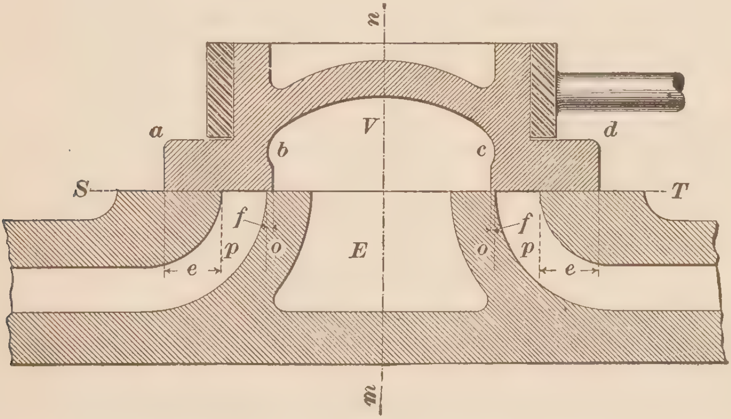

FIG. 5.

p, *p* are the **steam ports,** *o*, *o* the **bridges,** *E* the **exhaust port,** *S T* the **valve seat.** The flanges of the valve, *a b* and *c d*, are seen to be wider than the ports that they cover. Of this extra width, the parts *e*, *e* are called the

outside lap, and the parts *f*, *f* the **inside lap.** The valve is here shown in mid-position, i. e., the center line *n* of the valve coincides with the center line *m* of the exhaust port.

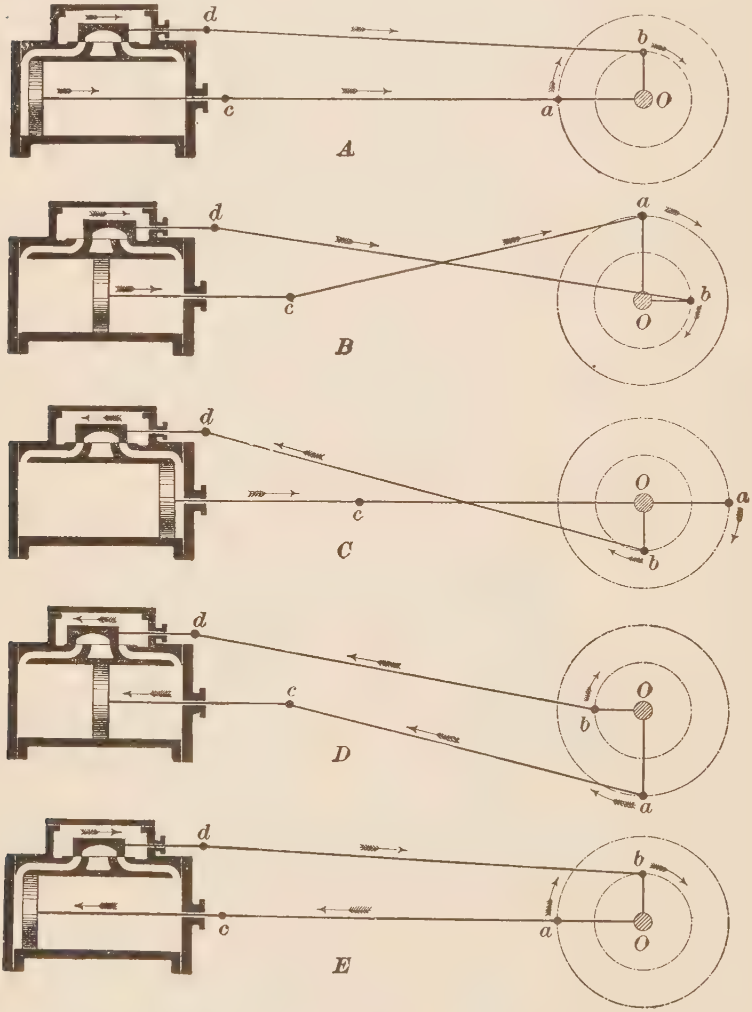

FIG. 6.

16. Action of Valve Without Lap or Lead.—Fig. 6 shows five diagrams that represent a **D** slide valve without lap or lead. *O a* represents the crank; *O b* the eccentric, which was shown to be equivalent to a crank; *a c* the connecting-rod; and *b d* the eccentric rod. It should be remembered that the relative sizes of some of the parts have been greatly exaggerated, particularly the radius of the eccentric circle and the amount of clearance. Diagram *A*, Fig. 6, represents the piston just on the point of beginning the forward stroke. The valve is moving in the direction of the arrow, and the outer edge is just about to admit steam to the left-hand port. As will be seen, the valve is in its central position, and, consequently, the line joining the center of the shaft and the center of the eccentric (this line will hereafter be called the **eccentric radius**) is vertical. All the parts are about to move in the direction of the arrows. Diagram *B* shows the positions of the parts when the crank has moved through 90° from its position in *A*. The piston is at the middle of its stroke, or very nearly there. It would be exactly at the middle of its stroke but for the fact that the connecting-rod makes an angle with the horizontal. It will be assumed here that this has no effect on the position of the piston. The valve has reached the extreme limit of its travel to the right and the eccentric radius *O b* is horizontal. The left steam port is fully opened for the live steam and the right steam port is fully opened for the exhaust. Another crank movement of 90° places the different parts as shown in diagram *C*. The piston has reached the end of its forward stroke; the valve is in its central position moving towards the left, and having just closed the left steam port and the right exhaust port, is about to open the right port for the admission of live steam and the left port for the release of exhaust steam. The piston has now traveled one full stroke. Diagram *D* shows the piston in its central position on the return stroke. The crank is in the position *O a*; the eccentric is horizontal, as represented by *O b*, and the valve is at the farthest point of its travel to the left, the right port being fully open for live steam and the left port fully open for exhaust. In the diagram *E* the piston has reached the extreme point of the return stroke, the piston rod, connecting-rod, and crank being all in one straight line; this also occurs in diagrams *A* (which is the same as *E*) and *C*. The valve has been moving to the right and is now in its central position, just on the point of admitting steam to the left port.

These diagrams show that, with no lap or lead, the steam is admitted to the cylinder for the full stroke of the engine; consequently, there can be no cut-off, and therefore no expansion of steam.

The following conclusion is now evident: *With an ordinary* **D** *slide valve, operated by an eccentric, there can be no cut-off, and therefore no expansion of steam, unless the valve has outside lap.*

17. Lap and Angle of Advance.—The effect of lap on the relative movement of the valve and piston, and also on the movement of the eccentric and crank, is clearly shown in Figs. 7 to 14. In these figures, the valve has both outside and inside lap, but no lead. These diagrams have been distorted, as was done in Fig. 6, in order that the eccentric radius might be long enough to show up well. In Figs. 7 to 14 the eccentric radius is three times as long as it should be for the amount of valve movement shown by the figure. The diameter of the crank circle is also a little greater than the stroke of the piston, for the same reason.

18. Diagram of Pressures in Cylinder.—In order to show the distribution of steam by the valve, a diagram has been drawn above and below each cylinder, those above being marked *M* and those below *N*. These diagrams are supposed to be drawn in the following manner: Imagine it to be possible to connect two small pipes to the piston, one on each side. Suppose each pipe to have a steam-tight piston working in it, the lower side of the pistons being subjected to the steam pressure in the cylinder and the upper side to the atmospheric pressure. Suppose, further, that there is a coiled spring on top of the piston; that a piston rod passes through the center of the spring; and that a pencil is attached to the end of the piston rod. If a pressure of 10 pounds is required to compress the spring 1 inch, it is evident that for every 10 pounds pressure in the cylinder, the pencil will move upwards 1 inch, and if it touched a sheet of paper, would mark a line on that paper. It will now be presumed that an arrangement like that just described is attached to the steam-engine piston, and that the pencil touches a sheet of paper that is held stationary. Then, when the steam-engine piston moves ahead, the pencil will make straight lines at heights corresponding to the steam pressure on the under sides of the little pistons, except when the pressure of the steam in the cylinder varies, in which case the pencil will move up or down, according as the pressure increases or diminishes.

19. Having made these suppositions clear, let QX, Figs. 7 to 14, represent the line that the pencil would trace if there were a perfect vacuum in the cylinder; i. e., QX is the line of no pressure; also let AB represent the line the pencil would trace if the pressure in the cylinder was just equal to that of the atmosphere, and QY the line of no volume. Then the point Q represents no volume and no pressure. Finally, let $1D$ represent the volume of clearance.

20. **Angle of Advance.**—Consider Fig. 7. The piston is represented as just beginning the forward stroke and the valve as just commencing to open the left steam port, both moving in the same direction, as shown by the arrows. If the valve had no outside lap, the position of the eccentric center would be at e, but on account of the lap, the valve had to be moved ahead of its central position in order to bring its edge to the edge of the port. To accomplish this, the eccentric center has been moved from e to b, Ob being the position of the eccentric radius. The angle bOe that the eccentric radius makes with the position it would be in if there were no lap or lead is called the **angle of advance.**

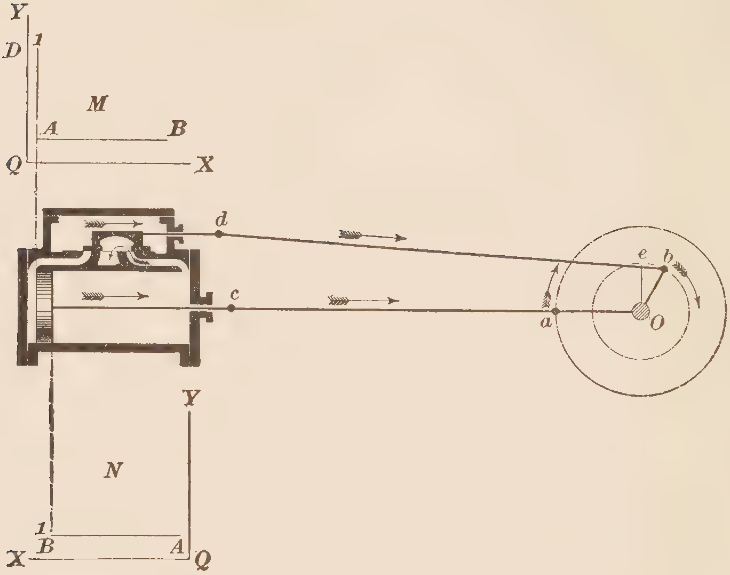

Fig. 7.

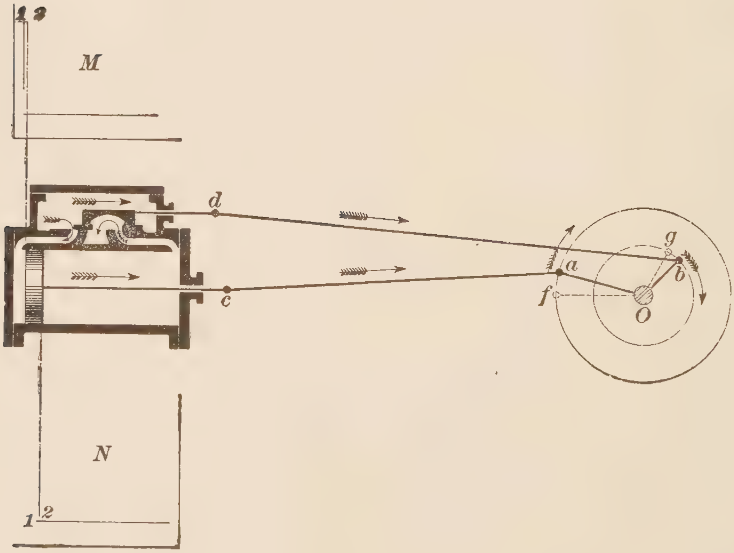

Fig. 8.

21. Back Pressure.—Assume that the piston and valve have moved a very small distance, just sufficient to admit steam to fill the clearance space on the left of the piston, so that the steam acts on the piston at full boiler pressure. If the length of the line *A 1* represents the boiler pressure (gauge), the pencil that registers the pressure on the left side of the piston will be at *1*. The steam on the right side of the piston is flowing (exhausting) into the atmosphere through the exhaust port, as shown by the arrow. As the size of the exhaust port is limited by practical considerations, the exhaust is not perfectly free, and there is a slight pressure on the exhaust side of the piston in addition to the atmospheric pressure. This is termed **back pressure.** Therefore, in the diagram *N*, let *1* be the position of the second pencil; then, *1 B* is the back pressure.

22. Exhaust Port Fully Open.—Fig. 8 shows the position of the piston and valve when the exhaust port is fully open. The crank has moved from the position *O f* (shown by dotted line) to *O a* and the eccentric center from *g* to *b*. Steam is entering the head end of the cylinder and exhausting at the crank end. The pencils have moved from *1* to *2* on both diagrams *M* and *N*.

23. Valve at the End of Its Stroke.—In Fig. 9, the piston has advanced far enough to enable the valve to reach the end of its stroke and open the port its full width. The crank and eccentric have moved to the positions *O a* and *O b*, the dotted lines showing their last position in Fig. 8. The eccentric radius is horizontal, and any further movement of the crank will cause the eccentric to travel in the lower half of its circle and make the valve move back. In the diagrams *M* and *N*, the pencil has traced the lines *2–3*.

24. Valve on Return Stroke, Steam Port Partly Closed.—Fig. 10 shows the piston still farther advanced on its stroke and the valve as having its inside edge in line with the outside edge of the exhaust port. The left end of

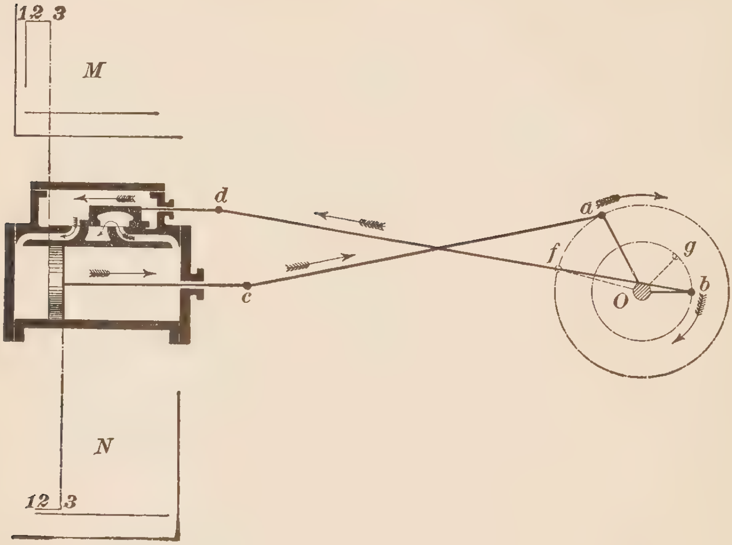

FIG. 9.

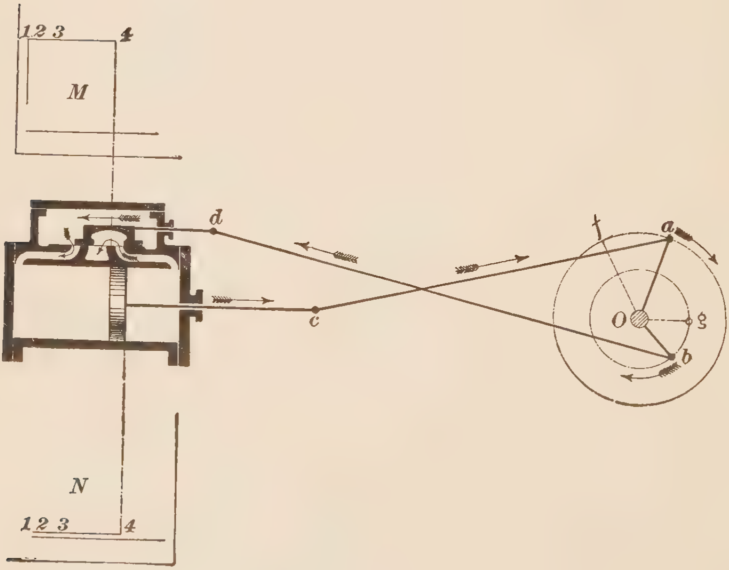

FIG. 10.

the valve has partially closed the steam port. The amount of advancement of the crank and eccentric from their last positions is shown by their distances from the dotted lines. The pencils have traced the lines *3–4* on the diagrams *M* and *N* during this movement of the piston from the last position.

25. Point of Cut-Off.—Fig. 11 marks one of the most important points of the stroke. Here the valve has closed the steam port, i. e., cut off the steam, and from here to the end of the stroke the steam in the cylinder expands. This is called the **point of cut-off.** The exhaust port is now partially closed. The crank and eccentric have moved through the angles indicated. During this movement, the pencils have traced the lines *4–5*.

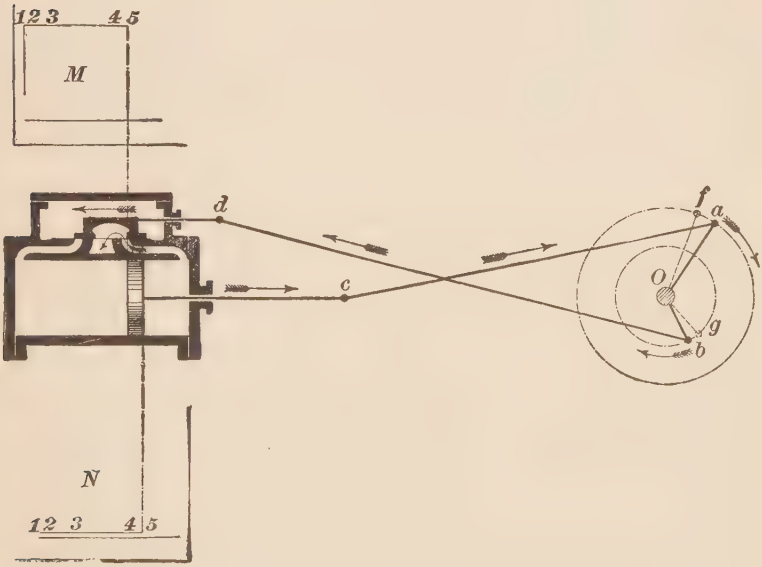

FIG. 11.

26. Point of Compression.—Fig. 12 shows another very important valve position. Here the inside edge of the valve closes the exhaust port, and from now to the end of the stroke, the steam in front of the piston is compressed. This point in the stroke is called the **point of compression.**

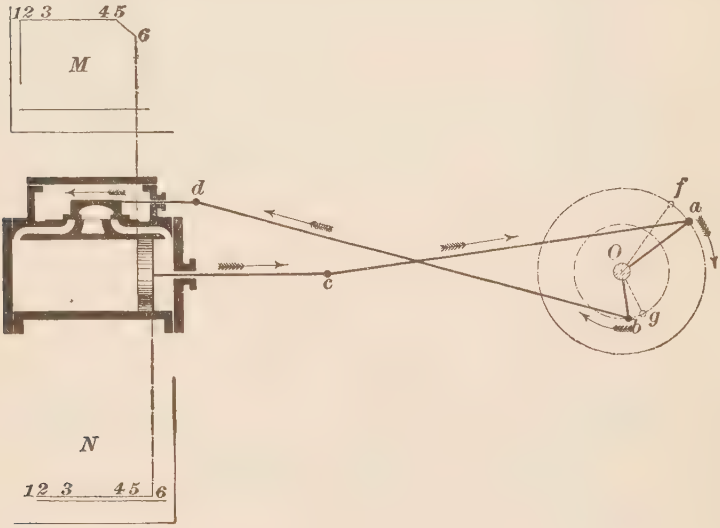

FIG. 12.

In the diagrams *M* and *N*, the lines *5–6* are traced by the pencils. The line *5–6* on the diagram *M* is an expansion line, the pressure falling as the piston moves ahead.

27. Point of Release.—In Fig. 13, the piston has advanced far enough to cause the left inside edge of the valve to be in line with the inside edge of the left port. The slightest movement of the valve to the left will open the left port so that the steam in the left end of the cylinder will pass into the exhaust port. This point of the stroke is called the **point of release.** The work done by expansion theoretically ends here, although, on account of the limitation in the size of the ports, there will still be a slight amount of work done by expansion, owing to the inability of the steam to escape instantly. During this last movement of the piston, the pencils trace the lines *6–7* on the diagrams *M* and *N*. On the diagram *M* the line *6–7* is a continuation of the expansion line *5–6*, while in the diagram *N* it shows part of the compression line, the pressure rapidly increasing as the piston nears the end of the stroke.

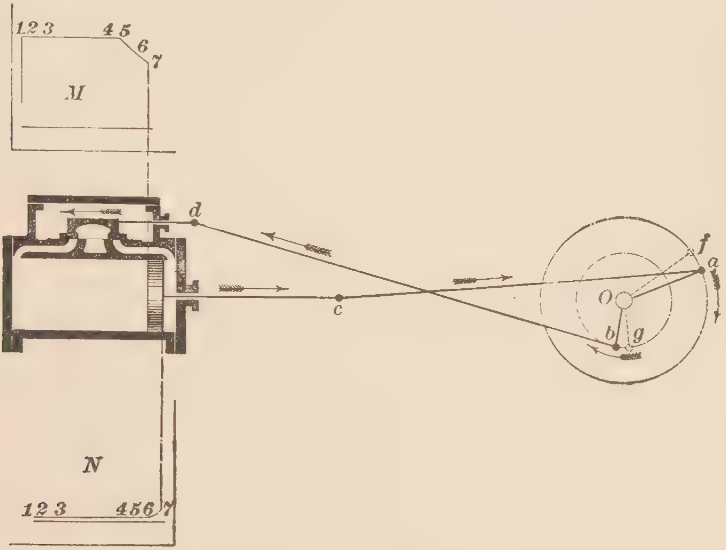

Fig. 13.

28. Piston at End of Stroke.—In Fig. 14, the piston has reached the end of its forward stroke and is about to begin the return stroke. The right outside edge of the valve is in line with the outside edge of the right port. The steam is exhausting from the head end of the cylinder, as shown by the arrows. The crank and eccentric are both diametrically opposite their positions in Fig. 7. In the diagrams *M* and *N*, the pencils have traced the lines *7–8*. *M* shows that the pressure has fallen very rapidly from *7* to *8*, while in *N* it has risen from *7* to *8*. The very slightest movement of the piston to the left will admit steam to the crank end of the cylinder and cause the pencil to rise to the point *1′*.

During the return stroke the above-described actions of the steam will be repeated, the pencils tracing the dotted lines on the diagrams *M* and *N* in Fig. 14 and the exhaust going through the left port and the steam through the right port. As the process is so nearly like the preceding, the diagrams have not been drawn, but the student should follow the valve through the different positions and note the effects on the diagrams. To assist him in this, the corresponding points have been numbered as in the foregoing figures.

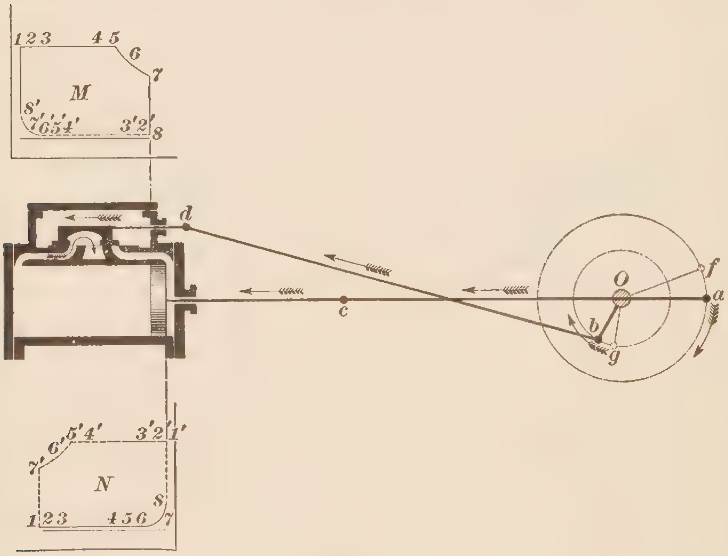

FIG. 14.

29. Effects of Lap.—A study of Figs. 7 to 14 should show the effects caused by varying the lap. Thus, in Fig. 11, it is evident that if the outside lap had been less, the valve would not close the left port when its center was in the position shown; consequently, the piston must move farther ahead before the valve can move back far enough to close the port. This, of course, makes the cut-off take place later in the stroke and shortens the expansion. It is likewise evident that if the valve had more lap, this extra lap would extend beyond the port when the center of the valve was in the position shown. Therefore, the valve would cut off earlier in the stroke and the expansion would be lengthened. Hence, *increasing the outside lap means an earlier cut-off and an increasing expansion, while decreasing the outside lap means a later cut-off and a diminished expansion.*

30. Considering the inside lap, it is evident from Fig. 12 that if the inside lap had been less, the exhaust port would not have closed so soon, and consequently the compression would have been later; had the inside lap been greater, the compression would have begun earlier. Fig. 13 shows that with a diminished inside lap, the release would begin earlier, while with an increased inside lap, the release would have taken place later in the stroke. *Increasing the inside lap causes the compression to begin earlier in the stroke and causes the release to take place later. On the other hand, diminishing the inside lap causes the compression to begin later and the release to take place earlier in the stroke.*

31. Lead.—In Fig. 7 the piston is just commencing the forward stroke and the valve is just about to uncover the left steam port. Most engineers, however, prefer to have the port a little open when the piston is at the end of the stroke. That is, the valve, instead of being just at the edge

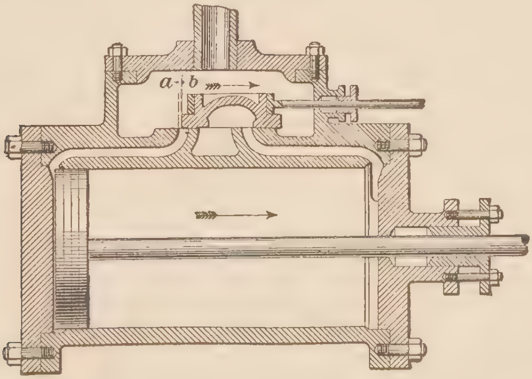

FIG. 15

of the port, as shown in Fig. 7, is moved $\frac{1}{16}$ inch or $\frac{1}{8}$ inch to the right, so that the clearance space is filled with fresh steam before the piston begins its stroke. A valve with lead is shown in Fig. 15. Here the piston is at the end of the stroke and the port is open a distance *a b*. This distance *a b* is the **lead.**

Since, when a valve has lead, it is moved farther to the right than in the position shown in Fig. 7, it is evident that the center *b* of the eccentric must also be moved a little farther to the right, Fig. 7. That is, to give a valve lead, the angle of advance must be increased.

32. Position of the Eccentric.—When the plain slide valve has neither lap nor lead, as in the skeleton diagrams, Fig. 6, it was shown that the eccentric must make an angle of 90° with the crank. Further, when the engine "runs over," as in Fig. 6, the eccentric is *ahead* of the crank. That is, following the direction of the arrows, the eccentric *b* reaches any point on its circle a quarter of a revolution before the crank *a* does. Referring now to Figs. 7 to 14, it is seen that when the valve has lap (or lap and lead), the angle *a O b* between the crank and eccentric is greater than 90°. Following the direction of the arrows, it is seen, however, that the eccentric *b* reaches, say, the lowest point on the circle earlier than the crank *a* reaches the lowest point on its circle. That is, the eccentric is *ahead* of the crank, as in the above case.

Take now the case of an engine that "runs under," as shown in Fig. 16. The crank is in position *a* and is about to move downwards. Now, the eccentric cannot be in the position *O b′*, for then it would move the valve to the left.

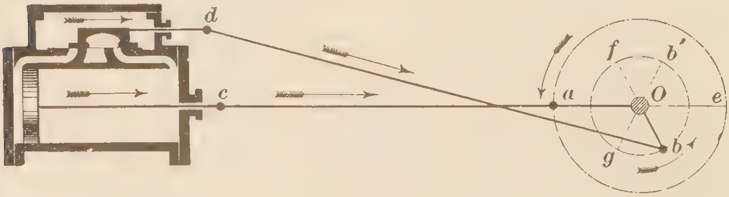

FIG. 16.

It cannot be opposite, in the position $O\,g$, for in that case, the valve would not be far enough to the right. It must be in the position $O\,b$. An inspection of the diagram shows that, following the direction of the arrows, the eccentric is set *ahead* of the crank, and the angle between the crank and eccentric is $a\,O\,b = 90° +$ the angle of advance.

Hence, for the ordinary slide valve, the following general direction applies: *When the valve rod and eccentric rod move in the same direction, the eccentric is set* **ahead** *of the crank, and the angle between the crank and eccentric is 90° + the angle of advance.* This law is true whether the engine runs "over" or "under."

33. Rocker-Arms.—It frequently happens that the eccentric cannot be so located on the shaft (in the direction of its length) that the eccentric rod and valve stem will be in the same straight line. It can never be done when the valve is on top of the cylinder without inclining the valve seat, now very seldom done, and with the valve on the side of the cylinder, other considerations, such as the location of the flywheel, may interfere. In such cases as this, a lever or rocker-arm may be used.

An example is shown in Fig. 2. It is perfectly evident that when the eccentric rod *11* moves to the left, the valve rod *13* will also move to the left, being compelled to do so by reason of the rocker-arm *12*. It is also plain that the amount of horizontal movement of the valve rod will be the same as it would be if the eccentric were attached directly to the valve rod, thus getting rid of the rocker-arm. The reason for using the rocker-arm in this case is that the eccentric-rod axis and valve-stem axis are not in the same straight line, the eccentric then being thrown too far to the left by the main bearing R'. The valve seat could, in this case, have been placed farther from the center of the cylinder, so as to bring the axes of the two rods in line. This, however, would have made the steam and exhaust ports that much longer. Since it is considered an advantage to have ports as short as possible, a rocker-arm was used.

34. Again, it is sometimes desirable to make the throw of the eccentric less than the valve travel. This may be accomplished by the use of a rocker-arm, as shown in Fig. 17. This rocker is pivoted at g and rotates about that point as a center. The valve rod is joined to the rocker at the end e and the eccentric rod is joined at d, a point between e and g.

Then, the eccentric throw must be smaller than the valve travel by the ratio $gd : ge \left(= \frac{gd}{ge}\right)$. For example, suppose the valve travel to be 4 inches, the distance $gd = 12$ inches, and $ge = 15$ inches. Then, the throw of the eccentric $= 4$ inches $\times \frac{gd}{ge} = 4 \times \frac{12}{15} = 3.2$ inches.

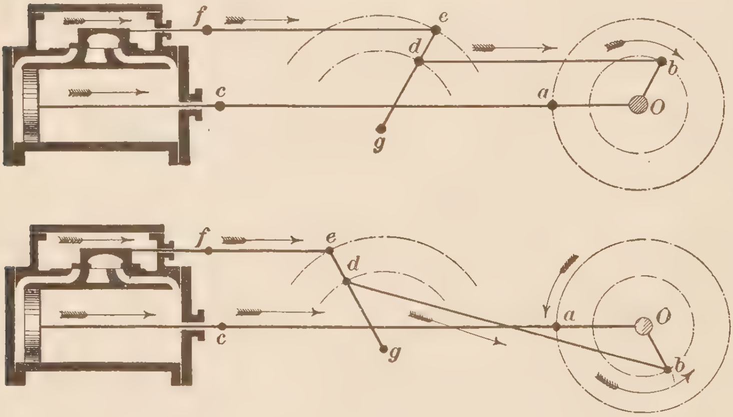

FIG. 17.

When the rocker is arranged as in Fig. 17, whether the engine runs over, as in the upper figure, or under, as in the lower figure, the valve rod and eccentric rod move in the same direction. Consequently, by the direction previously given, the eccentric is set 90° + angle of advance *ahead* of the crank.

35. It is often convenient to pivot the rocker near the center, as shown in Fig. 18. Here the points e and d, where the valve and eccentric rods, respectively, join the rocker, lie on opposite sides of the pivot g. As before, we have the

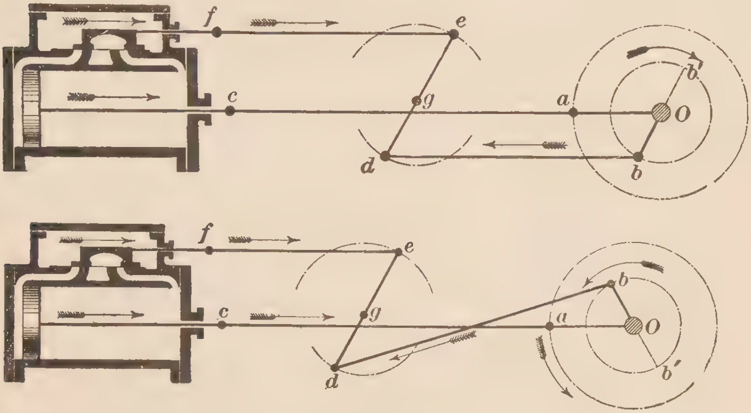

FIG. 18.

proportion — throw of eccentric : valve travel $= gd : ge$, or throw of eccentric = valve travel $\times \frac{gd}{ge}$.

It is easily seen, however, that when the rocker is pivoted near the center, as in Fig. 18, the valve rod and eccentric rod move in opposite directions. Consequently, to give the valve the proper motion, the eccentric rod must at all times move in a direction exactly opposite the direction of a rod attached as shown in Fig. 17. This can only be accomplished by placing the eccentric exactly opposite the position shown in Fig. 17. That is, instead of placing the eccentric 90° + angle of advance ahead of the crank, it must be placed 90° − angle of advance behind the crank. We have, therefore, the general direction: *In the case of the plain slide valve, if a rocker is so pivoted as to make the valve rod and eccentric rod move in opposite directions, the eccentric must be placed* **behind** *the crank, and the angle between the two is 90° − angle of advance.*

36. Direct and Indirect Valves.—A slide valve is said to be **direct** when it opens the left port upon moving to the right and closes it by moving to the left. A valve is said to be **indirect** when it opens the left steam port by moving to the left and closes it by moving to the right.

The plain slide valve already described is a direct valve. It opens the left port by moving to the right, admits steam past the outside edge, and exhausts it past the inside edge.

37. The **piston valve** shown in Fig. 19 is an example of an indirect valve. It consists of a hollow cylinder sliding in a cylindrical valve seat. The ports *P*, *P* extend clear around the valve. The steam is admitted into the central chamber *A* and the exhaust steam escapes out of the two ends *B*, *B*. As shown in the figure, the piston is just about to start to the right and the valve is moving to the left, thereby uncovering the left steam port and allowing the steam to enter past its inside edge. The valve is, therefore, indirect. To give a larger admission, steam also passes into the center of the valve through the channel *C* and thence into the left port. The exhaust steam meanwhile escapes directly through the right steam port into the chamber *B*.

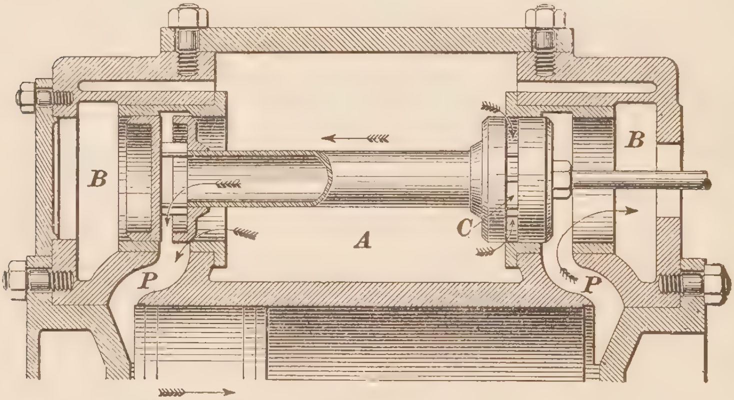

FIG. 19.

Attention is called to the fact that a piston valve is not necessarily an *indirect* valve; piston valves are often made as *direct* valves. In the latter case their action is exactly the same as that of the ordinary plain slide valve.

38. Eccentric Positions With Indirect Valves.—It is plain that the direction of motion of an indirect valve is precisely opposite that of a direct valve. Hence, as before explained, the eccentric must be set exactly opposite the position it would have were a direct valve used. We have, then, the following direction for the position of the eccentric: *When an indirect valve is used, set the eccentric* **behind** *the crank and make the angle between them equal 90° – the angle of advance. If a rocker is used that makes the valve rod and the eccentric rod move in opposite directions, set the eccentric* **ahead** *of the crank and make the angle between them equal to 90° + the angle of advance.* This rules applies whether the engine runs "under" or "over."

39. Table of Eccentric Positions.—The position of the eccentric relative to the crank for both the direct and indirect valves, direct and reversing rocker-arms, is given in the table. A rocker of the character shown in Fig. 17 will be called a *direct* rocker. One that changes the direction of the motion, as in Fig. 18, will be called a *reversing* rocker.

ECCENTRIC POSITIONS.

	Kind of Valve.	Kind of Rocker-Arm.	Angle Between Crank and Eccentric.	Position of Eccentric Relative to Crank.
I	Direct	Direct	90° + angle of advance	Ahead of crank
II	Direct	Reversing	90° – angle of advance	Behind crank
III	Indirect	Direct	90° – angle of advance	Behind crank
IV	Indirect	Reversing	90° + angle of advance	Ahead of crank

The above table may be applied equally well whether the engine runs over or runs under. It is simply necessary to remember that to set the eccentric ahead of the crank is to set it so that it reaches a given point in its revolution before the crank reaches the same point in its revolution. For example, in Fig. 16, suppose the engine to run under, as shown by the arrow. Then, the eccentric *O b* is set ahead of the crank *O a*, because it will reach the line *O e* before the crank will. If the eccentric were in the position *O f*, it would be behind the crank, because the crank would reach *O e* first. If, now, the engine should be supposed to "run over," then, if the eccentric were in position *O b* or *O g*, it would be behind the crank; if in position *O f* or *O b'*, it would be ahead of the crank.

DISTURBANCE OF CUT-OFF BY THE CONNECTING-ROD.

40. In Fig. 20, let *a b* represent the path of the center of the wristpin and *c d* the circle described by the center of the crankpin. Let the diameter of the circle *f g* equal the throw of the eccentric. (This is shown greatly exaggerated.) Assuming the crank to be in the position *O c*, that is, on the interior dead center, the length of the line *a c* will represent the length of the connecting-rod. We shall assume that the angle of advance is 20°; further, that the slide valve is set so as not to have any lead. *O e*, then, is the position of eccentric when crankpin is at *c*. Now, let the crankpin move in the direction of the arrow *x*; that is, let the piston commence its forward stroke. Since the valve has no lead, the slightest movement of the crankpin in the direction of the arrow will cause the valve to open the left steam port. When the eccentric has reached the position *O g*, the valve has moved to its farthest position to the right, and any

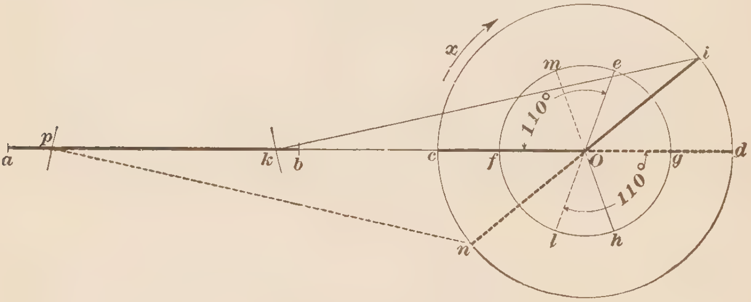

FIG. 20.

further movement of the crank will cause the valve to begin to close the steam port. To close the steam port fully, the valve will have to move the same distance to the left that it moved to the right to uncover the port. From this it follows that the eccentric must move through the same angle to close the port that it moved through to open the port. Laying off the angle $g O h = g O e$, $O h$ will represent the position of the eccentric at the time cut-off takes place. Laying off the angle $h O i = c O e$, we find the corresponding crank position. From the point i (the center of the crank-pin) as a center, with a radius equal to the length of the connecting-rod (the length of the line $a c$), describe an arc intersecting the line $a b$ at k; the point k will be the position of the center of the wristpin at the time of cut-off on the forward stroke. When the crank passes the exterior dead center, the right steam port will be opened; and at the moment that the crank occupies the position $O d$, the eccentric will be at $O l$; that is, $90° + 20° = 110°$ ahead of the crank. From what has previously been explained, it will be clear that the cut-off takes place on the return stroke when the eccentric reaches the position $O m$. The corresponding crank position will be $O n$. From n as a center, with a radius equal to the length of the connecting-rod, describe an arc intersecting $a b$ at p, which will give the position of the center of the wristpin at the time of the cut-off on the return stroke.

41. It will be seen at a glance that the cut-off has taken place considerably later on the forward stroke than on the return stroke, since $k b$ is less than $a p$. From this we see that a valve having an equal lap and set so as to have an equal lead cannot cut off equally on the forward and return stroke. If the valve is set so that the cut-off will be equal, the lead will be unequal.

This is due to the use of a connecting-rod. As a general rule, it may be stated that the longer the connecting-rod, the less will be the difference in the points of cut-off; and the shorter the connecting-rod, the greater the difference.

The effects of the connecting-rod on the steam distribution of a simple slide valve may be summarized as follows: *It will cause the valve to cut off and release the steam, as well as close the exhaust port, later on the forward stroke of the piston than on the return stroke.*

FORMS OF SLIDE VALVES.

42. Double-Ported Valves.—The plain **D** slide valve, shown in Fig. 5, is largely used on small engines running at moderately slow speeds. When, however, an engine has a high piston speed, the plain **D** valve does not open the port fast enough to allow the steam to follow up the moving piston and keep up full pressure in the cylinder. To overcome this difficulty, various forms of double-ported valves, one of which is shown in Fig. 21, are used. In Fig. 21, each port *P* has two openings *C* and *D*. The valve is made with two passages *B*, *B* extending through it; these passages connect with the steam chest *A*. In the position shown in the figure, the valve is opening the left steam port. The steam enters the passage *C* past the edge of the valve and enters the passage *D* through the opening in the chamber *B*. In the meantime, the exhaust is escaping from the right

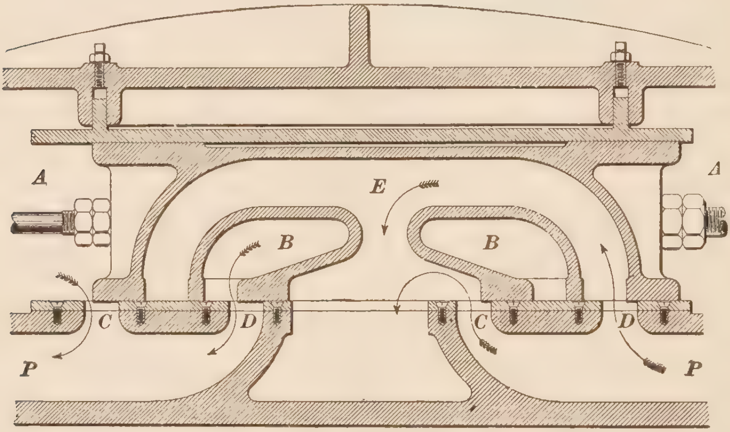

FIG. 21.

port into the chamber *E* beneath the valve. It is clear that, with the same travel, the double-ported valve gives double the opening to steam that the plain valve does. Otherwise, the two valves are alike in all respects.

43. The **Allen** or **Trick valve,** shown in Fig. 22, accomplishes the same object. The passage *A* is cast in the valve and extends clear through it. The shoulders *B*, *B* of the valve seat are so constructed that when the edge *m* of the valve is just even with the edge *l* of the port, the outer edge *p* of the passage *A* is just even with the edge *n* of the shoulder *B* at the other end of the valve seat. Now, when the valve moves a little to the right, into the position shown in the figure, steam enters the port directly between the edges *l* and *m*, as in the case of the ordinary valve. At the same time, the edge *p* of the passage has moved past the edge *n* of the valve seat; steam thus enters the passage *A* and finds there a direct path to the left steam port.

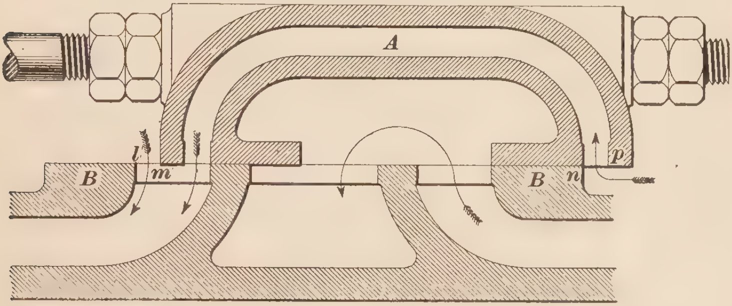

FIG. 22.

The piston valve shown in Fig. 19 is another example of a valve having a passage through it, by means of which the effective port opening for a given valve travel is doubled.

SETTING THE SLIDE VALVE.

44. Dead Centers.—Referring to Fig. 1, it is plain that when the piston *P* is at the end of its stroke at the end *h* of the cylinder, the crankpin *A* must lie at the point *m* in the crankpin circle. In this position the crank *O A* and connecting-rod *A B* lie in the same straight line. Likewise, when the piston is at the other end of the stroke, the pin *A* lies at the point *n*, and again the crank and connecting-rods are in the same straight line.

When the crank occupies either of these positions, the engine is said to be on its **dead center.** All the pressure of the steam on the piston is transmitted directly to the shaft *O*, because the reciprocating parts are in a straight line. Consequently, there is no tendency to turn the crank, and the engine cannot be started until turned into a different position. When the crank occupies the position *O m*, it is said to be on its **inner,** or **head-end,** dead center, and when it occupies the position *O n*, on its **outer,** or **crank-end,** dead center.

45. To Place the Engine on Its Dead Center.—It is sometimes necessary to place the engine exactly on its center

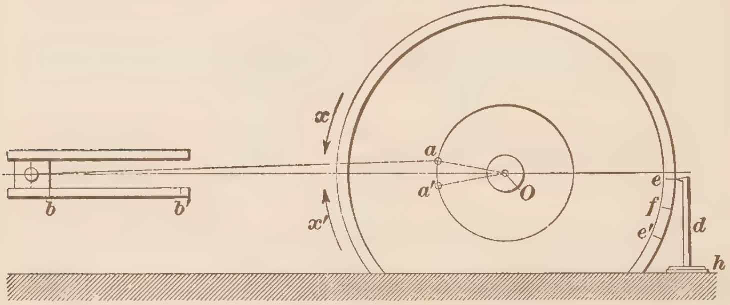

Fig. 23.

in order to set the valve. A common method of doing this is shown in Fig. 23.

When the crosshead is very near the end of its travel, make a mark *b* on one of the guides opposite the outer edge of the crosshead. Now turn the engine in the direction of the arrow *x* until the outer edge of the crosshead comes even with the mark *b*. While the engine is in this position, take a tram *d*, the length of which is about equal to the distance from the floor to the center of crank-shaft, place one end upon the floor (or, better, upon a solid block or part of the engine bed), and with the pointed end make a mark *e* upon the edge of the flywheel. The engine will probably not be exactly on the center; the crankpin will be, say, at a point slightly above the center. Now turn the engine in the direction of the arrow x' until the edge of the crosshead again comes even with the mark. The flywheel will have made nearly a complete revolution, and the crankpin will be at a', the same distance below the center that *a* was above it. Since the flywheel has made a little less than a full revolution, the mark *e* on the rim will not now be opposite the marking points of the tram, but the latter will make a new mark e' on the rim. Now, make a mark *f* half way between the marks *e* and e', and turn the wheel until the mark *f* comes opposite the point of the tram. The engine is then exactly on its dead center.

By taking another mark b' at the other end of the guide, the flywheel may be marked for the outer dead center. To insure accuracy, it is well to have both ends of the tram pointed. The lower point then fits into a prick-punch mark, made somewhere in the bed or foundation, and another punch mark on the rim determines the dead-center position.

46. Directions for Setting Slide Valve.— Put the engine on its dead center, place the valve on the seat and connect it with the eccentric rod. Shift the eccentric on the shaft until the valve has the desired lead. Turn the engine in the direction it is to run until it is on the other dead center. If the lead is the same as at the other end, the valve is correctly set; if it is not the same, the valve rod must be lengthened or shortened until the lead is the same at both centers. If, now, the lead is less than desired, shift the eccentric forwards a little on the shaft; if the lead is a little too great, shift the eccentric backwards.

After the valves are set and the engine is started, a pair of indicator diagrams should be taken. The diagrams will show any slight errors in the setting and corrections may be made accordingly.

CLEARANCE: REAL AND APPARENT CUT-OFF.

CLEARANCE.

47. Piston Clearance.—When the crank is on a dead center, the piston is always a short distance from the cylinder head; this allowance is made so that a slight change in the length of the connecting or piston rods will not cause the piston to strike the heads at the end of its stroke. It is also important to have a small space between the piston and head in which any small quantity of water in the cylinder may collect when the piston is at the end of its stroke; the incompressible nature of water would have the effect of breaking some part of the engine if there were no space in which it could collect. The short distance between the piston and the head when the piston is at the end of its stroke is called the **piston clearance.**

48. To measure the piston clearance at either end of the cylinder, first put the engine on its dead center for that end and make a mark on the guides corresponding to some convenient point of the crosshead. Next, disconnect the connecting-rod and push or pull the piston until it strikes the head. The distance of the chosen point on the crosshead from the mark made on the guides is the piston clearance for that end of the stroke.

49. The **clearance volume** or, simply, the **clearance,** is the volume of the space between the piston and cylinder head, when the piston is at the end of its stroke, plus the volume of the port leading to this space. Thus, in Fig. 15, the piston is at the end of its return stroke, and the clearance is the volume of the space between the piston and the left cylinder head plus the volume of the left steam port. In other words, the clearance may be defined as the volume of steam between the valve and the piston when the latter is at the end of the stroke.

50. Measuring the Clearance Volume.—The clearance volume of an engine may be found by putting the engine on a dead center and pouring in water until the space between the piston and the cylinder head and the volume of the steam port leading into it are filled. The volume of the water poured in is the clearance. Since water is likely to leak past the piston, some engineers advocate the use of a heavy oil for measuring the clearance volume.

51. Method of Expressing Clearance Volume.—The clearance volume may be expressed in cubic feet or cubic inches, but it is more convenient to express it as a percentage of the volume swept through by the piston. For example, suppose the clearance volume of a 12″ × 18″ engine is found to be 128 cubic inches. The volume swept through by the piston per stroke is $12^2 \times .7854 \times 18 = 2{,}035.8$ cubic inches. Then, the clearance is $\frac{128 \times 100}{2{,}035.8} = 6.3$ per cent. The clearance may be as low as $\frac{1}{2}$ per cent. in Corliss engines and as high as 14 per cent. in very high-speed engines.

52. Theoretically, there should be no clearance, since the steam that fills the clearance space does no work except during expansion; it is exhausted from the cylinder during the return stroke and represents so much dead loss. This is remedied to some extent by compression. If the compression were carried up to the boiler pressure, there would be very little, if any, loss, since the steam would then fill the entire clearance space at boiler pressure, and the amount of fresh steam needed would be the volume displaced by the piston up to the point of cut-off, the same as if there were no clearance. It is not practicable to build an engine without any clearance, because it is necessary to allow for lost motion and adjustment in the joints of the connecting-rod and because it is also necessary to allow for the formation of water in the cylinder due to the condensation of steam, particularly when starting the engine. As water is practically incompressible, some part of the engine would be broken when the piston reached the end of its stroke, provided there were no clearance space to receive the water; usually the cylinder heads would be blown off. Neither is it practicable to compress to boiler pressure, as a general rule, for that causes too great strains on the engine. Automatic cut-off, high-speed engines of the best design, with shaft governors, usually compress to about half the boiler pressure and have a clearance of from 5 to 14 per cent. Engines that do not have a high rotative speed, say not over 100 revolutions per minute, have very little compression and very small clearance. Such are the Corliss and other releasing-gear engines.

REAL AND APPARENT CUT-OFF.

53. It is customary, in speaking of the point of cut-off, to say that the engine cuts off at $\frac{1}{2}$ stroke, $\frac{1}{4}$ stroke, etc. By this is meant that the steam is cut off when the piston has completed $\frac{1}{2}$ or $\frac{1}{4}$ of its stroke, as the case may be. For example, if the stroke is 48 inches and the steam is shut off from the cylinder when the piston has moved 18 inches, the cut-off is $\frac{18}{48} = \frac{3}{8}$. The cut-off thus spoken of is the **apparent cut-off.**

54. The **real cut-off** takes account of the clearance space. It is the ratio between the volume of steam in the cylinder and clearance space when the piston is at the cut-off point and the volume of steam in cylinder and clearance space when the piston is at the end of the stroke. For example, let the volume of steam between the valve and piston when the latter is at cut-off be 4 cubic feet. Suppose that when the piston is at the end of its stroke the volume of steam in the cylinder and clearance space is 9 cubic feet. Then, the real cut-off is $\frac{4}{9}$.

55. The **relation between the apparent and real cut-offs** may be shown graphically as follows: Let the length AB, Fig. 24, represent the stroke of the engine. Suppose that the piston is moving in the direction of the

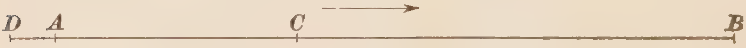

FIG. 24.

arrow and that the steam is cut off when the piston has reached the point C.

Then, according to the above, the apparent cut-off is $\frac{A\,C}{A\,B}$. It is clear that, since $A\,B$ represents the stroke of the piston, it will also represent, to some scale, the volume swept through by the piston. Now, to the same scale, lay off $A\,D$ equal to the volume of the clearance. Then, from the above definition, the real cut-off is $\frac{D\,C}{D\,B} = \frac{A\,C + A\,D}{A\,B + A\,D}$. Let s represent the apparent cut-off, k the real cut-off, and i the clearance expressed as a per cent of the stroke. Then, in Fig. 24, $s = \frac{A\,C}{A\,B}$ and $i = \frac{A\,D}{A\,B}$.

Rule 1.—*To find the real cut-off, add the clearance volume, expressed as a per cent. of the stroke, to the apparent cut-off, expressed in per cent., and multiply the sum by 100; divide the product by 100 plus the clearance volume, in per cent.*

Or,
$$k = \frac{(s + i) \times 100}{100 + i}.$$

Example.—In a 13″ × 18″ engine, the steam is cut off when the piston has moved over 8 inches of its stroke. The clearance is 8 per cent. of the volume displaced by the piston. Find the apparent cut-off and real cut-off.

Solution.—The apparent cut-off is $\frac{8 \times 100}{18} = 44.4$ per cent. Applying rule **1**, the real cut-off is found to be $\frac{(44.4 + 8) \times 100}{100 + 8} = 48.5$ per cent. Ans.

56. The **ratio of expansion,** also called the **number of expansions,** is the ratio between the volume of steam in the cylinder and clearance when the piston is at the end of its stroke and the volume in the cylinder and clearance when the piston is at the cut-off point. That is, in Fig. 24, the ratio of expansion is $\frac{D\,B}{D\,C}$. Since $\frac{D\,B}{D\,C} = \frac{1}{\frac{D\,C}{D\,B}} = \frac{1}{k}$, it follows that the ratio of expansion is the reciprocal of the real cut-off. For example, if the volume of steam behind the piston when at the end of its stroke is 15 cubic feet and when at cut-off is 5 cubic feet, the real cut-off is $\frac{5}{15} = \frac{1}{3}$. The ratio of expansion is $\frac{15}{5} = 3$; in ordinary language, the steam would be said to have 3 expansions.

When the *real* cut-off is given in per cent., the ratio of expansion is found by dividing 100 by the real cut-off in per cent. Thus, if the real cut-off is 25 per cent., the ratio of expansion is $\frac{100}{25} = 4$.

Illustrative Example.—Let it be required to find the clearance, the actual cut-off, and the ratio of expansion of a $12'' \times 24''$ engine under the following conditions: When the engine is on its center, the water from a vessel which with the water weighed 5 pounds was poured into the end of the cylinder. After pouring in just enough water to fill the clearance space, the vessel and water were weighed and found to weigh $1\frac{3}{4}$ pounds; consequently, the weight of water poured out of the vessel was $5 - 1\frac{3}{4} = 3\frac{1}{4}$ pounds. The weight of 1 cubic inch of water is .03617 pound. The number of cubic inches poured into the cylinder is, therefore, $\frac{3.25}{.03617} = 89.85$ cubic inches, nearly, which is the volume of the clearance space. The area of the 12-inch piston is $12^2 \times .7854 = 113$ square inches, very nearly; consequently, the piston displacement is $113 \times 24 = 2{,}712$ cubic inches. The clearance volume is, therefore, $\frac{89.85 \times 100}{2{,}712} = 3.31$ per cent. of the piston displacement; in other words, we say that the clearance is 3.31 per cent.

The cut-off takes place when the piston has moved 15 inches of its stroke. The apparent cut-off is, therefore, $\frac{15 \times 100}{24} = 62.5$ per cent. of the stroke. In accordance with rule **1**, the real cut-off is $\frac{(62.5 + 3.31) \times 100}{100 + 3.31} = 63.7$ per cent.

In accordance with Art. **56**, the ratio of expansion is $\frac{100}{63.7} = 2$, very nearly.

EXAMPLES FOR PRACTICE.

1. What is (*a*) the clearance volume in cubic inches, and (*b*) the clearance in per cent. of a 16″ × 24″ engine, if $7\frac{1}{4}$ pounds of water are required to fill the clearance space?

Ans. (*a*) 200.4 cu. in.
(*b*) 4 per cent.

2. If the engine in example 1 cuts off when the piston has made 6 inches of its stroke, what is (*a*) the apparent cut-off, (*b*) the real cut-off, and (*c*) the ratio of expansion?

Ans. (*a*) $\frac{1}{4}$, or 25 per cent.
(*b*) 27.9 per cent.
(*c*) 3.59, nearly.

THE BILGRAM VALVE DIAGRAM.

57. Graphic Method of Determining the Effect of Change in Proportion of Valves.—The action of the valve of a plain slide-valve engine when operated by an eccentric can be readily analyzed by means of a diagram that has been designed by Mr. Hugo Bilgram. This diagram is extremely useful not only in the analysis of an existing slide-valve gear, but as it also exhibits in a graphic form the effects of any change in the proportions of the valve, it is invaluable in the design of a new valve.

The valve diagram and its application to an existing valve gear is shown in Fig. 25. In a case of this kind, the outside and inside lap of the valve, the travel of the valve, and the stroke of the engine are known; and the lead, if not known, may be assumed. With these data, the amount that the steam ports are opened (the port opening), the point of cut-off, the point of release, the point of exhaust closure, and the angle of advance of the eccentric can be determined.

58. To any convenient scale, draw on the line *a b*, with *o* as a center, the semicircle *a d b* having a radius equal to that of the crank (one-half the stroke). About *o* as a center and with a radius equal to one-half the valve travel, describe the semicircle *a′ o′ b′*. Draw a line *g h* parallel to *a b* and at a distance from it equal to the lead. With a radius equal to the outside lap of the valve and with a center *o′* on the semicircle *a′ o′ b′* describe a circle *r* that is tangent to gh. The position of the center o' is easiest found by trial. About o' as a center and with a radius equal to the inside lap of the valve, describe a circle s. Next draw the straight lines od, ol, and om tangent, respectively, to the outside lap circle r and the inside lap circle s. Through o and o' draw the straight line of. From the points of intersection d, l, and m of the lines od, ol, and om with the semicircle adb, drop perpendiculars, as dk, lp, and my on the straight line ab. About o as a center describe an arc tangent to the outside lap circle and intersecting ab in t.

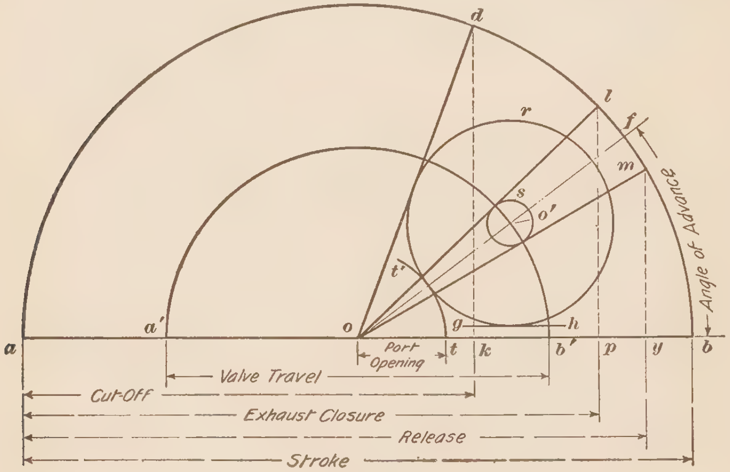

FIG. 25.

In the diagram just drawn, the distance ot represents the port opening to the same scale to which the diagram was drawn; the distance ak shows the piston movement up to the point of cut-off; the distance ay shows the piston movement up to the point of release, and the distance ap shows the piston movement up to the point where the exhaust port is closed, i. e., up to the point where compression begins. The angle fob is the angle of advance.

59. When the valve has no inside lap, no inside lap circle can be drawn; in that case, drop a perpendicular from the intersection point f of the line $o f$ with the semicircle $a d b$ to the line $a b$. The distance between a and the point of intersection of this perpendicular with the line $a b$ will then represent the piston movement up to the points of compression and release.

60. A careful study of the diagram will show the effects of changes in the valve proportions and valve setting. Thus, suppose that, the valve proportions remaining the same, it has been decided to give more lead. Then, the lead line $g h$ being at a greater distance than before from $a b$, the center o' will be higher up and farther to the left; in consequence, the intersection points d, l, and m will also be to the left of their former positions and the intersections of perpendiculars dropped from these points on $a b$ will be nearer a than before. In other words, the increase of lead causes the different events to take place earlier. Since o' will occupy a different position than formerly, the angle $f o b$ (the angle of advance) will now be greater; this shows that, with the valve proportions remaining the same, increasing the lead is impossible without increasing the angle of advance.

61. Suppose that the inside lap is *decreased*, all other valve proportions and the lead remaining the same. Then, since the inside lap circle s will be smaller, the point l will be nearer f than before, and, consequently, the intersection of a perpendicular dropped from l on $a b$ will be to the *right* of p; that is, the exhaust closure will take place later. The intersection point m will also be closer to f, and, obviously, the intersection of a perpendicular dropped from it on $a b$ will be to the *left* of y; that is, the release will take place earlier.

Suppose that the outside lap is *increased* in order to get an earlier cut-off and that the lead is to remain as before. Then, owing to the greater diameter of the circle r, its center o' will be farther to the left, and, consequently, the tangent lines $o d$, $o l$, and $o m$, and their perpendiculars $d k$, $l p$, and $m y$ will also move to the left, thus indicating that all the events will take place earlier.

62. Assume that while the outside lap was increased, no change was made in valve travel. Then, since the lap circle *r* is larger than before, it follows that *o t* will be smaller; that is, an increase of outside lap not accompanied by an increase of the valve travel will cause a decrease in the port opening. From this it is evident that in order to keep the port opening constant, an increase of outside lap must be accompanied by an increase of the valve travel.

By studying in the manner indicated in the preceding articles the effect on the valve diagram of any change in the valve proportions, in the angle of advance, etc., the effects on the steam distribution can be noted easily.

63. In designing a new valve for an engine, the port opening, the point of cut-off, and the stroke of the engine are known; the lead must be fixed upon. With these data, the valve proportions, the valve travel, and the angle of advance are readily determined by means of the Bilgram valve diagram.

On the line *a b* and from *o* as a center, describe the semicircle *a d b* to represent the path of the crankpin, using any convenient scale. Draw the lead line *g h* parallel to *a b* and at a distance equal to the lead from it. With a radius equal to the port opening, describe an arc *t t′* about *o* as a center. On *a b* lay off *a k* equal to the desired cut-off. At *k* erect a perpendicular, and from its point of intersection *d* with the semicircle *a d b* draw the straight line *d o*. Now, by trial, find the radius and the position of the center *o′* of a circle that will be tangent to the lines *o d* and *g h* and tangent to the arc *t t′*. The radius of this circle represents the outside lap required, while the distance of the center *o′* from *o* represents one-half the valve travel. By drawing a straight line, as *o f*, through *o* and *o′*, the angle of advance is determined. The question of how much, if any, inside lap to give is one that each designer must answer for himself, remembering that the giving of inside lap makes release later and exhaust closure earlier.

SIZE OF STEAM PASSAGES.

64. The average practice of steam-engine builders is to proportion the steam passages so that the steam will flow at a velocity of 6,000 feet per minute through the main steam pipe, 6,000 feet per minute through steam ports that are relatively long and tortuous, as the steam ports of plain slide-valve engines, 7,500 feet per minute through very short and direct steam ports, 4,000 feet per minute through the exhaust ports, and also through the exhaust pipe.

65. With these velocities as a basis, the following rules for proportioning the steam passages have been deduced. In the case of steam pipes and exhaust pipes, the commercial size of pipe whose area is nearest the calculated area should be selected. In case either the steam pipe or the exhaust pipe, or both, is very long, say above 200 feet, it may be advisable to select a pipe one size larger than the one whose area is nearest the calculated area. In case the steam pipe is but poorly or not all protected by covering, the use of a larger size of pipe is especially necessary.

66. Rules and Formulas for Calculating the Sizes of Steam Passages.—

Let a = area of steam port in square inches;
b = area of exhaust port and pipe in square inches;
c = area of steam pipe leading to engine;
d = piston speed in feet per minute;
e = area of piston in square inches.

Rule 2.—*To find the area of the steam pipe leading to an engine or pump, multiply the area of the piston by the piston speed in feet per minute and divide the product by 6,000.*

Or,
$$c = \frac{d\,e}{6{,}000}.$$

EXAMPLE.—A 14″ × 36″ engine is to run 100 revolutions per minute. What size should the steam pipe be?

Solution.—The piston speed in feet per minute is $\frac{36}{12} \times 100 \times 2 = 600$ feet. The area of the piston is $14^2 \times .7854 = 153.938$, say 154 square inches. Applying rule **2,** we get

$$c = \frac{154 \times 600}{6,000} = 15.4 \text{ square inches.}$$

The nearest commercial size of pipe is 4½ in. nominal diameter, whose internal area is 15.939 sq. in. Ans.

67. **Rule 3.**—*To find the area of the exhaust pipe or exhaust port for an engine or steam pump, multiply the area of the piston by the piston speed in feet per minute and divide the product by 4,000.*

Or, $$b = \frac{d\,e}{4,000}.$$

Example.—What should be the area of the exhaust port and what should be the size of the exhaust pipe of the engine mentioned in the example given in Art. **66**?

Solution.—Applying the rule just given, we get

$$b = \frac{154 \times 600}{4,000} = 23.1 \text{ square inches}$$

as the area of the exhaust port. Since a 6-inch pipe has an actual internal area of 28.889 square inches, while the next smaller commercial size of pipe, viz., 5-inch, has an area of but 19.99 square inches, experienced engineers would select the 6-inch pipe, in order not to cramp the exhaust. Ans.

68. **Rule 4.**—*To find the area of the steam port, multiply the area of the piston by the piston speed in feet per minute; divide the product by 7,500 if the port is short and direct, and by 6,000 if the port is long and tortuous.*

Or, $$a = \frac{d\,c}{7,500} \text{ for a short port,}$$

and $$a = \frac{d\,e}{6,000} \text{ for a long port.}$$

Example.—What should be the area of the steam port for the engine given in the example of Art. **66,** if the port is short?

Solution.—Applying rule 4, we get

$$a = \frac{154 \times 600}{7,500} = 12.32 \text{ sq. in.} \quad \text{Ans.}$$

THE ROTARY ENGINE.

69. Since the time of Watt, it has been the aim of many inventors to produce an engine in which the piston has a rotary motion, thus dispensing with the connecting-rod and crank. Innumerable designs have been proposed and patented, many of which have been actually tried; except for special service where economy in the use of steam is but a minor consideration, they have all proved commercial failures. It is a very simple matter to design a rotary engine that will turn (run); it is an entirely different matter, however, to have a rotary engine develop in constant and extended service a horsepower on the same steam consumption as a reciprocating steam engine.

70. Rotary engines have been constructed in a great variety of forms, many of which can only be characterized as freaks. The remainder usually consist either of a rotary piston of some suitable form, bearing against a rolling, sliding, or swinging abutment, or a design of interlocking pistons similar to that found in the Root blower. Abutments, no matter how carefully designed, made, and fitted, cannot be kept steam-tight for any length of time, and generally will cause bad steam leakage into the exhaust in a short time; rotary engines of the interlocking piston pattern either commence to leak badly after short service or are very noisy. Some rotary engines are valveless and very simple, but extremely costly to operate. Some have eccentric pistons; these rapidly wear the cylinder walls and bearings owing to the difficulty of counterbalancing them properly for high speeds. On the whole it can be safely stated that the rotary engine, owing to inherent irremovable constructive difficulties, will never be a serious commercial competitor of the reciprocating engine.

THE STEAM TURBINE.

71. Two eminent engineers, Mr. Parsons and Mr. Laval, have developed a steam engine working on an entirely different principle than the ordinary steam engine. Instead of making use of the pressure of steam, they utilize the kinetic energy contained in a mass of steam moving with a very high velocity, jets of steam impinging against the blades or vanes of a wheel fitted inside of a suitable casing and thus rotating the wheel at a high speed. This kind of an engine, from its similarity to the turbine, is called a **steam turbine.** A limited number of steam turbines are in use, the latest designs giving an economy about equal to that of the reciprocating steam engine.

THE INDICATOR.

INDICATORS AND REDUCING MOTIONS.

CONSTRUCTION OF INDICATOR.

GENERAL CONSTRUCTION.

1. The **indicator** is an instrument that can be readily applied to a steam engine for the purpose of obtaining a diagram of the pressures in the cylinder. It is made in a variety of forms that differ, however, only in minor details;

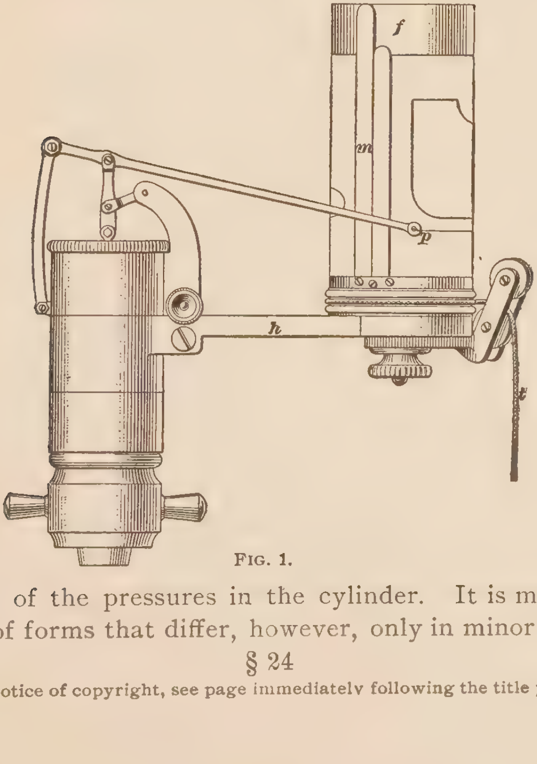

FIG. 1.

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the general principles involved in all will readily be understood by reference to Fig. 1, which shows the general appearance of an indicator, and Fig. 2, which shows one in

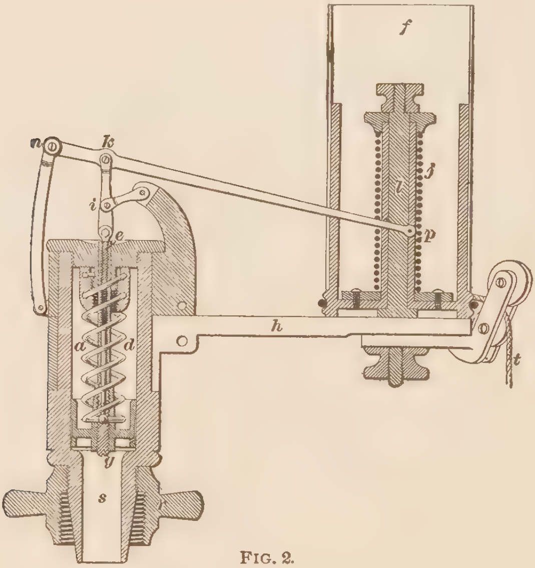


FIG. 2.

section. The instrument consists essentially of a cylinder *a*, Fig. 2, containing the piston *g* and the spring *d*. By turning a cock connected to the small pipe to which the indicator is attached, steam may be admitted to, or shut off from, the cylinder of the indicator at pleasure. When steam is admitted through the channel *s*, its pressure causes the piston *g* to rise. The spring *d* is compressed and resists the upward movement of the piston. The height to which the piston rises should then be in exact proportion to the pressure of the steam, and as the steam pressure rises and falls, the piston must rise and fall accordingly.

2. To register this pressure, a pencil might be attached to the end of the piston rod *e*, the point of the pencil being made to press against a piece of paper. It is desirable, however, to restrict the maximum travel of the piston to about ½ inch, while the height of the diagram may advantageously be 2 inches or more. To obtain a long pencil movement combined with a short travel of the piston, the pencil is attached at p to the long end of the lever $n\,k\,p$. The fulcrum of the lever is at n. The piston rod is connected to it at k through the link $i\,k$. The pencil motion is thus $\frac{p\,n}{n\,k}$ times the piston travel. This ratio $\frac{p\,n}{n\,k}$ is, for most indicators, either 4, 5, or 6. The point p is forced to move in a vertical straight line by the arrangement of the links and joints i, e, n, and k, forming what is called a **parallel motion.**

DETAILS OF INDICATOR.

3. The Spring.—The height to which the piston will rise under a given steam pressure depends upon the stiffness of the spring. Indicators are usually furnished with a number of springs of varying degrees of stiffness, which are distinguished by the numbers 10, 20, 30, 40, etc. These numbers indicate the pressure per square inch required to raise the pencil 1 inch and are called the **scale of the spring.** Thus, if a 40 spring is used, a pressure of 40 pounds per square inch raises the pencil 1 inch, and the vertical scale of the diagram is, therefore, 40 pounds per inch; that is, the vertical distance in inches of any point on the diagram from the atmospheric line, multiplied by 40, gives the gauge pressure per square inch at that point. The scale of the spring chosen should not be less than one-half the boiler pressure. For example, we would choose a 40 spring for a steam pressure of 75 pounds per square inch.

4. The Paper Drum.—The indicator must not merely register pressures, but it must register them in relation to the position of the piston. To accomplish this object, a cylindrical drum f, Figs. 1 and 2, is provided. This drum can be revolved on its axis l by pulling the cord t, which is coiled around it. When the pull is released, the spring j rotates the drum back to its original position. If, now, the cord t be attached to some part of the engine that has a motion proportional to the motion of the piston, the motion of the drum will also be proportional to the motion of the piston.

CONNECTING INDICATOR TO ENGINE CYLINDER.

5. To attach the indicator to the engine, a hole is drilled in the clearance space of the cylinder and tapped for a ½-inch nipple. If this hole is in the top of the cylinder, the indicator cock may be screwed directly into it, or, if more convenient, a nipple and coupling may be used. If the cylinder is tapped at the side, a nipple and elbow may be used so as to bring the indicator into a vertical position; since, however, it is desirable to keep the connections to the indicator as short and direct as practicable, some engineers prefer to omit the elbow and attach the indicator in a horizontal position. The indicator is attached directly to the cock by the nut *r*, Fig. 2, which wedges the conical projection *s* of the indicator tightly into the cock and thus prevents leakage of steam. On account of the resistance offered by the pipe and elbows to the flow of steam to the indicator, it is preferable to have an indicator at each end of the cylinder, but if that is not convenient, one indicator may be connected with both ends of the cylinder by means of a three-way cock.

6. Most cylinders of the better class of engines are now provided with bosses having holes tapped in them for the convenient application of the indicator; in many old engines, however, no special provision for the indicator has been made. In such cases care must be taken to drill holes that will not be covered with the piston when it is near the end of the stroke. If the hole cannot be tapped directly into the clearance space, a passage must be chipped to the clearance space in order that the steam can reach the indicator.

7. The indicator connection should never be on the side of the cylinder directly opposite to the steam ports; the current of entering steam would strike against the opening leading to the indicator, and the pressure shown by the diagram would thereby be considerably increased. With many engines that have not been provided with special indicator attachments, it will be found that holes for this purpose may be conveniently drilled and tapped in the cylinder heads.

INDICATORS FOR SPECIAL PURPOSES.

8. Gas-Engine Indicators. — An indicator specially adapted to gas engines, and one that is sometimes applied to steam engines, pumps, and hydraulic machinery when high pressures are used, is shown in Fig. 3. The cylinder has two bores *a* and *b*. The larger bore *a* is ½ square inch in area (the size usually employed when testing a steam engine), and the area of the smaller bore *b* is ¼ square inch. The piston *c* is fitted to the smaller bore and is that used when indicating a gas engine or a very high pressure steam engine. It gives but half the movement of the pencil given by the larger piston used in *a*; so, if the spring used is stamped 40, the calculations, when using the smaller piston, must be made as if an 80-pound spring had been used. The pencil movement is also of special design, the moving parts being stronger and more rigid than those used on regular steam-engine patterns. With the regulation piston of ½ square inch area in chamber *a*, the indicator may be used for steam engines with ordinary pressures.

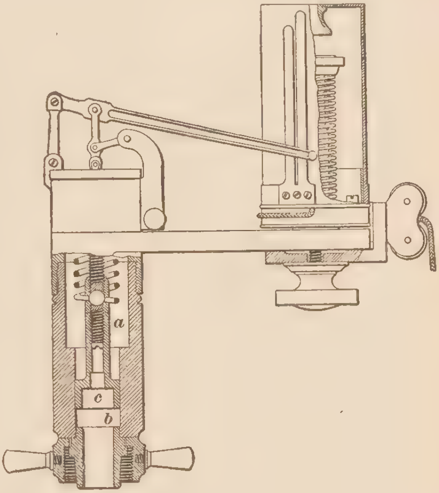


FIG. 3.

9. Ammonia Indicators.—For use on the ammonia cylinders of refrigerating machines, it is preferable to use a special indicator the working parts of which are made of steel instead of brass, because ammonia has no effect on steel, but rapidly corrodes brass. In case it is not possible to procure an ammonia indicator, an ordinary steam-engine indicator will answer the purpose, provided the piston is removed after every set of cards is taken and both cylinder and piston are wiped dry and well covered with oil. This will prevent the ammonia gas from attacking the portions of the indicator made of brass.

REDUCING MOTIONS.

PANTOGRAPH MOTIONS.

10. Purpose of a Reducing Motion.—The motion of the paper drum is nearly always taken from the crosshead. However, since the stroke of the crosshead is longer than the circumference of the drum, it is necessary to arrange a form of mechanism some point of which will copy to a reduced scale the stroke of the piston. Such a mechanism is called a **reducing motion.**

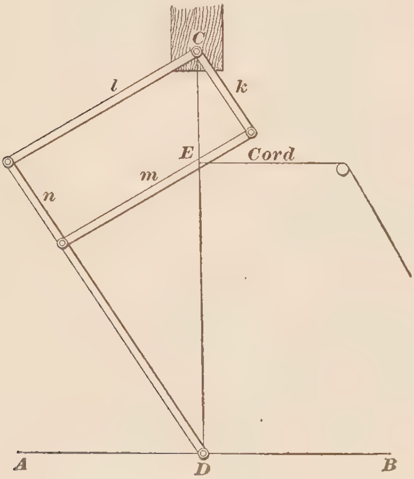


FIG. 4.

11. The **pantograph,** Fig. 4, is an excellent form of reducing motion. It consists of four links joined together in the form of a parallelogram. One of the links *n* is prolonged and is pivoted at the end to the crosshead *D*. The opposite corner of the parallelogram is pivoted to the fixed point C. The cord is attached to the point E on the link m, which point must be on the straight line connecting C and D. $A B$ represents the length of the stroke. Letting h represent the length of the indicator diagram, we have the following proportions: $A B : h = C D : C E$, or $\frac{A B}{h} = \frac{C D}{C E}$.

12. The **lazy tongs,** Fig. 5, is a modified form of the pantograph that is much used as a reducing motion. It consists of a series of bars joined together in such a manner as to form a flexible frame. The joint A of the frame is attached to any convenient stationary point on the engine or its surroundings and B is attached to the crosshead. The bar C is provided with a number of holes, in one of which is placed a pin c to which the indicator cord is attached. The bars D and E to which C is joined are provided with a series of holes, and C may be placed in any position in which its ends are attached to holes similarly located in the two bars. For each position of C, one of the holes in it will be in line with the two joints A and B at the extremity of the frame; this is the hole in which the pin c is to be placed for that position of C.

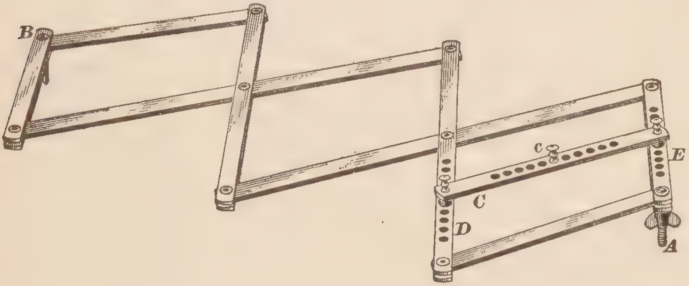


FIG. 5.

13. The ratio of the length of the diagram to the length of the stroke is equal to the ratio of the distance $A c$ to the distance $A B$; this is true for any distance between the points A and B. To find the correct position of the bar C

for a given length of card, when the length of stroke is known, set the points *A* and *B* at some convenient distance apart; multiply this distance by the desired length of the card and divide the product by the length of the stroke; the quotient so obtained will be the distance from *A* at which to set the pin *c*, keeping *A* and *B* at the distance apart to which they were set at the beginning of the operation. A very convenient method of locating *C* is to make the distance *A B* equal to the length of the stroke and then locate *C* so that the distance *A c* is, as nearly as possible, equal to the desired length of the diagram. For example, let it be desired to take a diagram 3½ inches long from an engine having a stroke of 32 inches; make the distance *A B* 32 inches and then attach *C* to the holes in *D* and *E* that will make the distance *A c*, as near as may be 3½ inches.

14. The pantograph and lazy tongs are theoretically correct reducing motions; that is, the motion imparted to the indicator cord is exactly proportional to the motion of the crosshead. The point to which the cord is attached moves in a straight line parallel to the direction of motion of the crosshead. The fixed point of either the pantograph or lazy tongs may be at any place that will enable the cord to be led to the indicator in the shortest and most direct manner; it is not necessary, as is sometimes assumed, to locate the fixed point on a line equidistant from both ends of the stroke. In locating the point of attachment, however, it is important to guard against striking the joints of the frame at the ends of the stroke; neglect of this precaution may result in breaking the reducing motion. A disadvantage of the pantograph is the danger of lost motion due to wear in the joints.

SWINGING-LEVER REDUCING MOTIONS.

15. A lever with one end pivoted at some convenient fixed point and the other attached to the crosshead of the engine forms one of the simplest reducing motions; and if the device is correctly designed and carefully constructed, it can be made to give as accurate results as are obtainable in any way. Two common forms of swinging-lever reducing motions are those illustrated in Figs. 6 and 7. The form shown in Fig. 6 is called the **slotted swinging lever.**

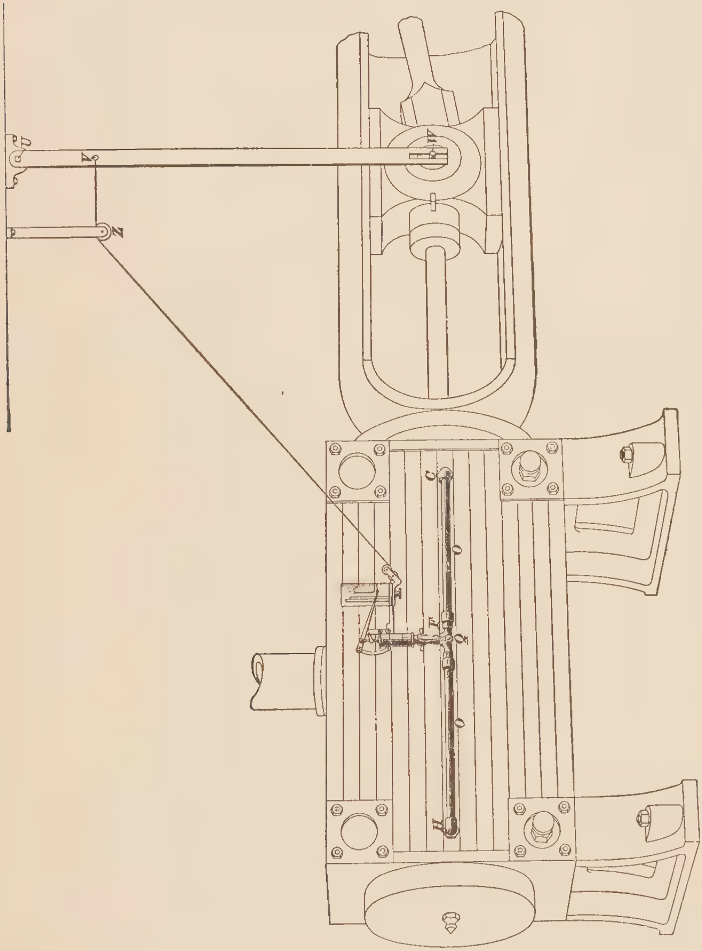


FIG. 6.

The lower end of the lever is slotted and fits over a pin in the crosshead; the other end of the lever is pivoted at a fixed point *U* and the cord is attached at *V*. The cord is guided by a pulley *Z* so that it will leave the point *V* in a direction parallel to the line of motion of the crosshead.

In the device shown in Fig. 7, the lever is connected to the crosshead by a link *W D*. The cord is attached to the circumference of a grooved sector *E V F*, called a **Brumbo pulley.** The center of the circle of which the sector forms a part lies in the center line of the lever pivot *U*. The sector may be attached directly to the lever or it and the lever may be keyed to a short shaft in such a manner that the cord can be led directly to the indicator. Instead of the sector, the cord may be attached to a pin, as in the motion shown in Fig. 6; in this case a guide pulley similar to the pulley *Z*, Fig. 6, would be required.

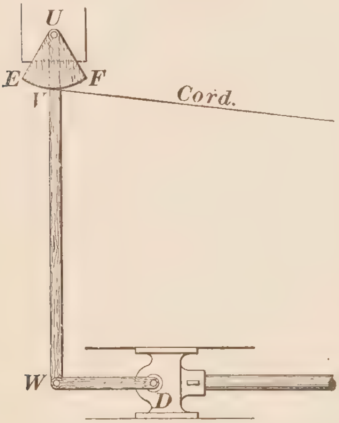


FIG. 7.

16. Error of Swinging-Lever Motions.—The types of swinging-lever motion illustrated in Figs. 6 and 7 are imperfect, from the fact that the motion imparted to the cord is not exactly proportional to the motion of the crosshead. In the slotted swinging lever, Fig. 6, the distance from the pivot *U* to the center of the pin in the crosshead is variable, while the distance from *U* to the point *V* to which the cord is attached is constant; in other words, the length of the long arm of the lever varies, while the length of the short arm remains constant. This results in a variation in the relative motions of crosshead and cord for different parts of the stroke, and the diagram obtained is, in consequence, distorted.

With the motion illustrated in Fig. 7, the link *W D* acts like a connecting-rod to transmit the straight-line motion of the crosshead to the end of the lever that moves in a circular arc; the link thus has an angular motion that has a disturbing effect on the ratio of the cord movement to that of the crosshead. The result is a diagram whose proportions are not perfect.

17. Methods of Reducing the Errors.—For ordinary work with the indicator, the amount of distortion with carefully made swinging-lever motions is not serious and may be ignored. To secure good results, the levers should always be suspended from such a point that when the crosshead is at the middle of its stroke, they will be perpendicular to its line of motion. The accuracy of the motion will, in general, be increased by increasing the length of the levers; for most purposes it will be sufficient to use a lever whose length is twice that of the stroke, and in some cases a lever even shorter than this is used. The accuracy of the motion with the connecting link, Fig. 7, is also increased by increasing the length of the link; for ordinary work, a link whose length is equal to one-third of the length of the stroke may be used. The lever in this motion should be so suspended that the extreme positions of *W* above and below the line of motion of the point *D* are about equal.

18. Theoretically Correct Motions.—The errors of the swinging-lever motions that were noted in Art. **16** can be neutralized and a theoretically correct reduction obtained by the methods illustrated in Figs. 8 and 9. In each of the figures the cord is attached to a sliding bar *S* whose line of motion is parallel to that of the crosshead. In Fig. 8 the bar *S* is provided with a pin that works in a slot in the swinging lever. By this arrangement the ratio of the distances from the pivot *U* to the pins *W* and *V* is constant for all positions of the crosshead, and the motion of the bar *S* is exactly

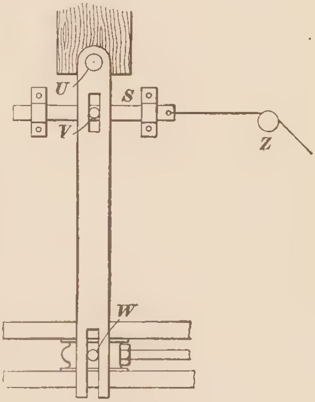


FIG. 8.

proportional to that of the crosshead. In Fig. 9 the bar S is connected to the swinging lever by a short link $V C$. In order to secure a theoretically correct reduction, the length of this link must be such that the ratio of the distance $V C$ to the distance $W D$ is equal to the ratio of $U V$ to $U W$, and the bar S must be so located that the center lines $V C$ and $W D$ of the two links are parallel. When these conditions are fulfilled and the points U, V, and W lie in the same straight line, the center of the joint C lies on the straight line joining U and D.

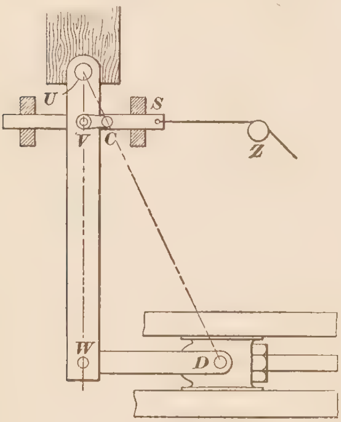


Fig. 9.

The cord must be guided by pulleys, as Z, so that it will leave the bars S, Figs. 8 and 9, in a direction parallel to the line of motion of the bars.

19. Rule for Proportioning Swinging-Lever Motions.—With any swinging-lever reducing motion, the ratio of the length of the diagram to the length of the stroke is equal to the ratio of the distance from the pivot to the point of attachment of the cord to the distance from the pivot to the pin by means of which the lever is connected to the crosshead; thus, with either of the motions shown in Figs. 6 to 9, let l represent the length of the diagram and L the length of the stroke, then $\frac{l}{L} = \frac{U V}{U W}$. In accordance with this principle, we have the following

Rule.—*To find the distance from the pivot at which to connect the cord, or to find the radius of the Brumbo pulley when the length of the stroke, the length of the diagram, and the distance from the pivot of the lever to the point where it is connected with the crosshead are known, multiply the length of the diagram by the distance from the pivot to* *the point of the lever at which it is connected with the crosshead and divide the product by the length of the stroke.*

Let $l =$ length of diagram;
$L =$ length of stroke;
$d =$ distance from pivot to point of attachment of cord (see $U\,V$, Figs. 6 to 9);
$D =$ distance from pivot to point where lever is connected to crosshead (see $U\,W$, Figs. 6 to 9).

Then, $$d = \frac{D\,l}{L}.$$

Example 1.—The stroke of an engine is 28 inches; the length $U\,W$ of the lever is 6 feet; what must be the distance $U\,V$ to give a diagram 3½ inches long?

Solution.—Applying the rule just given, we have

$$d = \frac{72 \times 3\frac{1}{2}}{28} = 9 \text{ in. Ans.}$$

Example 2.—In Fig. 7 find the radius $U\,V$ of the arc $E\,F$ in order that the diagram may be 3½ inches long, the stroke of the engine being 38 inches and the length $U\,W$ being 5 feet 5 inches.

Solution.—This example is solved in the same manner as the preceding one. The effective length of the lever is 5 feet 5 inches = 65 inches. Applying the rule, we have

$$d = \text{radius of arc } E\,F = \frac{65 \times 3.5}{38} = 6 \text{ in., nearly. Ans.}$$

REDUCING WHEELS.

20. Reducing wheels form a very convenient and theoretically accurate method of reducing the motion of the crosshead to the required value for the paper drum. These wheels are furnished in a variety of forms, some of which are designed to be attached directly to the indicator, while others are provided with means for clamping to some point on the engine bed. One of the latter type is illustrated in Fig. 10. The cord a is attached to a bar or rod fastened to the crosshead in such a manner that the cord will lead from it to the wheel in a line parallel to the line of motion of the crosshead, and the cord *b* is attached to the indicator drum. The smaller pulley can be removed and replaced by one of several others of different sizes. The proportions of the two pulleys can thus be varied so as to secure the desired length of diagram. Thus, if the stroke of the engine is 12 inches and the desired length of the diagram is 3 inches, the diameter of the larger pulley should be four times that of the smaller. The hub of the larger wheel contains a spring that is wound up when the cord *a* is unwound from the wheel by the outward motion of the crosshead; when the crosshead makes its return stroke, the spring turns the wheel and winds the cord on again.

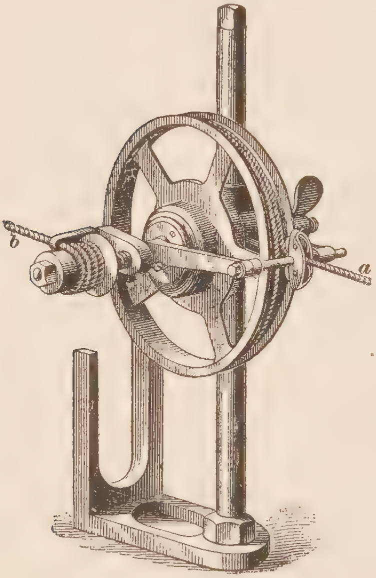


FIG. 10.

REDUCING MOTIONS FOR HIGH SPEEDS.

21. When an engine has a high rotative speed, the quick changes in direction of motion set up severe stresses in a reducing gear. In a pantograph or a swinging-lever motion, these stresses are likely to cause a springing of the parts that will distort the diagram and lead to erroneous results. The shocks and stresses also tend to wear the joints rapidly; the lazy-tongs motion, with its great number of joints and moving pieces, is on this account poorly adapted for high-speed work. A swinging lever, if made of stiff and light wood with joints bushed and neatly fitted, will give good results at nearly any speed of rotation at which it is practicable to run an engine. Instead of bushing the joints, they may be made adjustable as shown in Fig. 11. The end of the bar is split by a saw kerf passing through the center of the hole forming the pivot bearing and extending far enough into the bar to permit the two parts to be drawn up tight against the pivot by a wood screw *S*. Instead of the screw a somewhat more substantial method is to use a small bolt passing through the end of the bar. A joint made in this way and fitted to a turned pin, if well lubricated, gives the best satisfaction and lasts almost indefinitely.

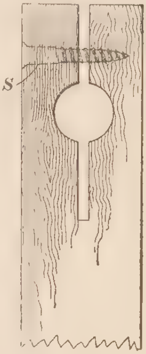


FIG. 11.

22. With reducing wheels the quick reversals of the direction of rotation that take place at high speeds make it necessary to use a stiff spring to overcome the inertia of the wheel. To reduce the inertia, the wheels are made as light as is practicable; with many reducing wheels, lightness is secured by the use of aluminum. The better class of reducing wheels can now be successfully used for nearly any speed of rotation likely to be met with.

INDICATOR CONNECTIONS.

23. Indicator Cords.—In order to transmit the motion from the reducing motion to the paper drum with as little loss or distortion as possible, it is necessary to use a cord that will stretch but little. To meet this requirement, indicators are generally supplied with a special braided cord that will give good results for most purposes. In the case of large engines, where long cords are required, the amount of stretch with the best cord obtainable is considerable and may result in a distortion that would be undesirable for accurate tests. For such cases a fine copper or steel wire may be used to advantage. It is always best to so arrange the reducing motion and indicator that the cord may be led to the paper drum in the shortest and most direct practicable line. When the cord is attached to a pin on the reducing motion, it must be guided so as to leave the pin in the line of its motion, as is illustrated in Figs. 6, 8, and 9; the use of guide pulleys should, however, be avoided as much as it is practicable.

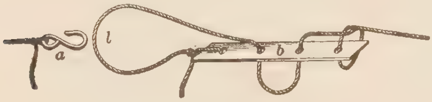


FIG. 12.

24. Stop Motions.—Various means are used to stop the motion of the paper drum when it is required to change the paper or when the indicator is not in use. A common method is to have a short cord attached to the paper drum with a hook *a*, Fig. 12, on the end; the cord from the reducing motion has a loop into which the hook may be fastened when it is desired to operate the paper drum. The length of the cord from the reducing motion can readily be made adjustable by the use of a loop *l*, formed as shown in Fig. 12. The thin strip of wood or metal *b* provides a very ready means of changing the length of the loop and of tying it securely in any position. To prevent the cord leading from the reducing motion from being thrown about and getting tangled when it is unhooked from the cord leading to the paper drum, it is well to have a rubber band fastened in a convenient position and provided with a hook into which the loop *l* may be secured. The elasticity of the rubber band can thus be made to keep the cord stretched and to prevent it from being tangled and broken.

25. Paper drums are sometimes provided with a stop motion that will hold them in place and prevent the cord from being wound on; this merely has the same effect as lengthening the cord, but is open to the objection that at high speeds the loose cord is apt to make trouble by flying about and getting caught. In addition to the stop motions above noted, indicator manufacturers have designed a number of very useful devices, some of which absolutely prevent any trouble with the cord and make it easy to start and stop the paper drum.

SPECIAL ATTACHMENTS.

26. Simultaneous Diagrams.—With an engine having two or more cylinders it is sometimes desirable to take a diagram simultaneously from each end of all the cylinders so as to get a record of what takes place in each cylinder at some particular time. It is difficult for a number of operators to apply the pencils of a set of indicators to the paper all at the same time; and, to overcome this difficulty, a number of devices have been invented by means of which the pencils of all the indicators can be simultaneously operated by one person. Of these devices the simplest and most successful is an electromagnet that is attached to the indicator. When a number of indicators are to be operated simultaneously, the electromagnets of all are connected by a wire; when a current is sent through the wire by pressing a button or key, each electromagnet pulls its pencil against the paper and holds it there until the circuit is opened.

INDICATOR DIAGRAMS.

DIRECTIONS FOR TAKING DIAGRAMS.

27. The makers of indicators furnish very complete instructions for the care and use of their instruments; these instructions should be carefully studied before attempting to use a new indicator or one with which the user is not thoroughly familiar. The following directions for taking diagrams apply to all makes of indicators: Before attaching the indicator to the engine, see that it is clean and in good working order. The piston should move freely. See that the joints of the various levers and links are oiled with fine oil and that they are slack enough to avoid friction, yet not so slack as to allow the pencil to shake. Adjust the pencil so that it just touches the paper and sharpen the point so that it makes a very fine light line. A heavy coarse line on a diagram indicates poor work.

Select a spring that will give a diagram about 1½ or 1¾ inches in height. If, upon trial, the spring chosen makes a wavy line, choose a stiffer one. A stiffer spring is required on a fast-running engine than on a slow-running engine when the steam pressure is the same. See that there is no backlash between the piston and spring.

Adjust the length of the cord so that the drum turns backwards and forwards without striking either of the stops at the end of the travel. When it touches one or the other of the stops, the cord is either too short or too long. If it touches both, the travel of the drum is too great, and the cord must be fastened to a point on the reducing motion having less travel.

Keep the drum moving only when taking diagrams. Unhook the cord before putting a paper on the drum. In putting on the card, see that it fits the drum without wrinkles, and fold back the projecting edges over the clips *m*, Fig. 1, so that they will not touch the pencil lever.

28. Before taking the diagram, turn on the steam a minute or so to warm the indicator; then press the pencil lightly on the paper long enough to take a single diagram. Shut the cock and again press the pencil to the paper. Since the indicator piston is then only subjected to atmospheric pressure, the pencil will make a straight line called the atmospheric line. Disconnect the cord and remove the card. Write on the card the scale of the spring used, the speed of the engine, and any other desired particulars.

If one indicator is used for both ends, first open the three-way cock to admit steam from one end. Take the diagram and open the cock to the other end, and take the diagram from that end. Then shut off the steam entirely and take the atmospheric line.

GENERAL FEATURES.

29. Purpose.—In actual practice the imperfections in the construction of the engine and the velocity at which the steam must flow through the pipes and ports combine to modify the pressures in the cylinder and, in consequence, the form of the diagram drawn by the indicator pencil. By a careful study of the peculiar features of the diagram, an experienced engineer is able to determine with a considerable degree of certainty the general type and condition of the engine and the circumstances under which the diagrams were taken. The following general outline of the characteristic features of diagrams taken under different conditions will enable the student to interpret most of the diagrams with which he will meet.

30. Points and Lines of the Diagrams.—In Figs. 13 and 14 are shown indicator diagrams from the crank end and head end of an engine. The different *points* of the stroke are plainly shown. They are as follows:

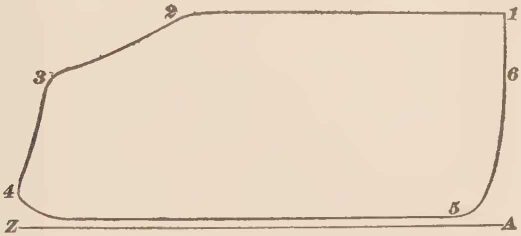


FIG. 13.

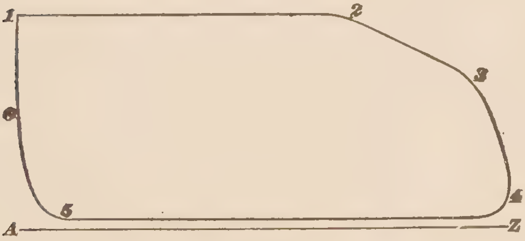


FIG. 14.

1 is the beginning of the stroke.
2 is the point of cut-off.
3 is the point of release.
4 is the end of the stroke.
5 is the point of compression.
6 is the point of admission.

The *lines* included between any two of these points have received special names, which are as follows:

6–1 is the admission line.
1–2 is the steam line.
2–3 is the expansion curve.
3–4–5 is the period of release.
4–5 is the back pressure line.
5–6 is the compression curve.
A Z is the atmospheric line.

31. The Vacuum Line.—It is sometimes desirable to have the vacuum line (line of no pressure) on the card also. The vacuum line may be drawn below and parallel to the atmospheric line. The distance between them will be $\frac{14.7}{\text{scale of spring}}$ inches. Thus, if the scale of the indicator spring is 30, the vacuum line lies $\frac{14.7}{30} = .49$ inch below the atmospheric line. When great accuracy is desired, the vacuum line should be located in accordance with the indication of the barometer. This is especially desirable when the engine is located at a great altitude above sea level. Then the distance between the atmospheric and vacuum lines may be found by multiplying the reading of the barometer in inches by .49 and dividing by the scale of the spring. For instance, if the barometer stands at 25 inches and the scale of the indicator spring is 30, the vacuum line should be drawn at a distance of $\frac{25 \times .49}{30} = .41$ inch from the atmospheric line.

32. Two Diagrams on a Single Card.—If but one indicator is used, the two diagrams are taken on the same blank, as shown in Fig. 15. With the diagrams placed one over the other, as shown, it is very easy to tell what is taking place in the cylinder at any point of the stroke. On the forward stroke the pencil of the indicator describes the line *A B C D* of the head diagram if the cock is opened to the head end, or it describes the line KLM if the cock is opened to the crank end. Likewise, the lines $GHJK$ and DEF are described during the return stroke.

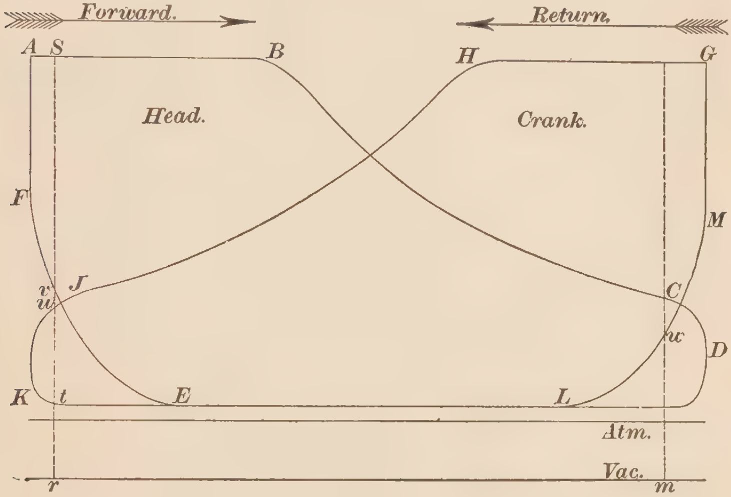


FIG. 15.

Suppose the piston is at a position corresponding to r on the forward stroke; the pressure (above vacuum) urging the piston forwards is rS, while the pressure resisting is rt. Hence, the net pressure on the piston is St. Suppose, now, that the piston is at r on the return stroke; the pressure at the right urging the piston on is ru, while the pressure on the left is rv. The net pressure is, therefore, uv, and is negative; or, in other words, the resistance is greater than the effort.

33. A double diagram of this character tells at a glance what is taking place at either end of the cylinder at any point of the stroke. Thus, when the piston is on the forward stroke, in the position corresponding to m, the steam in the head end is at the point of release C. Draw a line through m perpendicular to the vacuum line. C lies on ABC, and since KLM is described at the same time as ABC, the intersection of the line through C with the line *K L M* is the point corresponding to *C*. The intersection *w* is on the compression line; hence, when release occurs in the head end, compression is taking place in the crank end.

34. Effect of Type and Speed of Engine on Form of Diagram.—The form of a good diagram depends largely on the type of the engine, style of valve, and speed. What would be considered a good diagram from a locomotive or from a high-speed automatic engine would be considered very poor if taken from a Corliss engine. In general, a diagram taken from an engine with releasing gear of the Corliss type will be regular and show but little compression. The point of cut-off, release, and compression will be sharply marked. The diagram shown in Fig. 16 is what may be

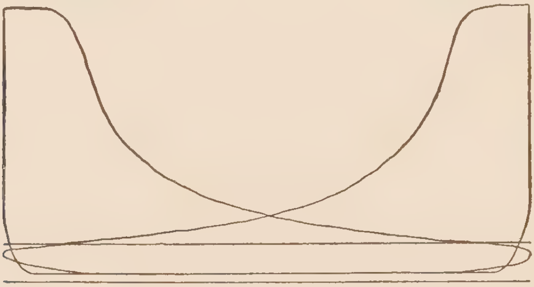


FIG. 16.

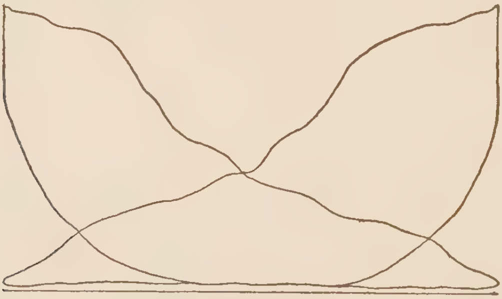


FIG. 17.

expected from this type of engine when the valves are correctly set and in good working order. The fact that the back-pressure line runs below the atmospheric line shows plainly that the engine the card was taken from was condensing. On the other hand; Fig. 17 shows the form of diagram that may be expected from an engine running at 250 to 300 revolutions per minute. On account of the high rotative speed, the lines are irregular, due to the inertia of the moving parts of the indicator. The compression is large, as it should be for engines running at a high speed. The point of cut-off is never very sharply marked.

It is readily seen how totally unlike are the diagrams shown in Figs. 16 and 17, yet each is considered as representing good practice.

FAULTS IN VALVE SETTING REVEALED BY DIAGRAMS.

FAULTS IN STEAM DISTRIBUTION.

35. Some of the most common faults revealed by the indicator diagram are given below. In the diagram following, Figs. 18 to 22,

1 is the admission.
2 is the cut-off.
3 is the release.
4 is the compression.

I. Admission may be too early.
II. Admission may be too late.
III. Cut-off may be too early.
IV. Cut-off may be too late.
V. Release may be too early.
VI. Release may be too late.
VII. Compression may be too early.
VIII. Compression may be too late.

36. Case I. Admission Too Early.—The effect on the diagram of a too early admission is shown in Fig. 18. It is seen that the admission line *1a*, instead of being straight and perpendicular to the atmospheric line, as in Figs. 13 to 16, slants backwards. With a single slide valve or piston valve, all the other events, cut-off, release, and compression, are also too early. The remedy is to shift the eccentric on the shaft so as to decrease its angular advance.

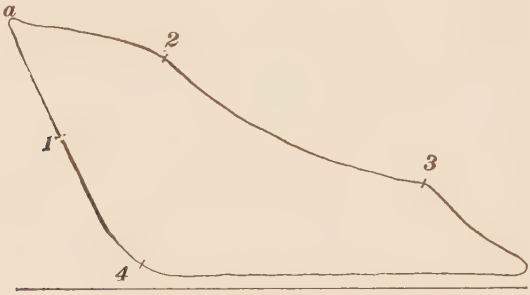


FIG. 18.

In the case of the Corliss engine, the admission may be too early, while the other points are not affected. The fault may then be remedied by adjusting the link rods so as to give the steam valves more lap, and it may not be necessary to shift the eccentric.

The effect of too early admission is to introduce an excessive resistance to the motion of the piston before it reaches the end of the stroke; in consequence, the piston must be pushed to the end of its stroke by the momentum of the flywheel acting through the crank and connecting-rod. The result is likely to be pounding at the crosshead, crank, and main bearing.

37. Case II. Admission Too Late.—In this case the admission line *1a* on the diagram slants forwards, as shown in Fig. 19. The remedy is to increase the angular advance until the admission line *1a* becomes perpendicular to the atmospheric line. With a single slide valve in the case of a too late admission, the other events, as *2*, and particularly *3* and *4*, are also too late.

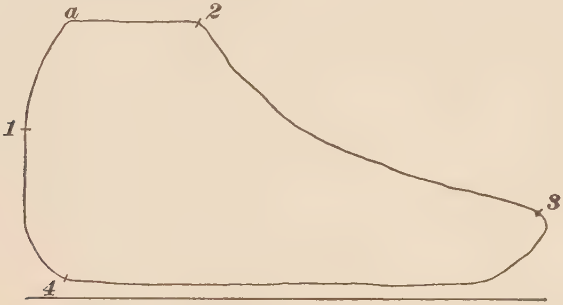


FIG 19.

In case the engine has a Corliss or other releasing gear, the admission may be made earlier by reducing the lap of the admission valves.

The effect of late admission on the running of the engine is not generally as noticeable or severe as is too early admission. It is, however, generally desirable to give the valves enough lead to have the clearance space filled with steam at the boiler pressure just as the piston begins its stroke.

38. Case III. Cut-Off Too Early.—See Fig. 20. Here the steam expands until its pressure is less than the back

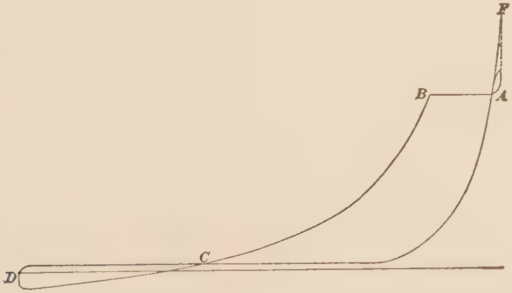


FIG. 20.

pressure; in consequence, the expansion line crosses the back-pressure line, as shown at *C*, and forms a loop. This effect is often observed in automatic cut-off engines working under a light load. It causes a reversal of the pressures on the piston that may result in pounding. The great range in pressure also has a bad effect on the steam consumption.

39. Case IV. Cut-Off Too Late.—See Fig. 21. Here it will be noticed that the terminal pressure is very high.

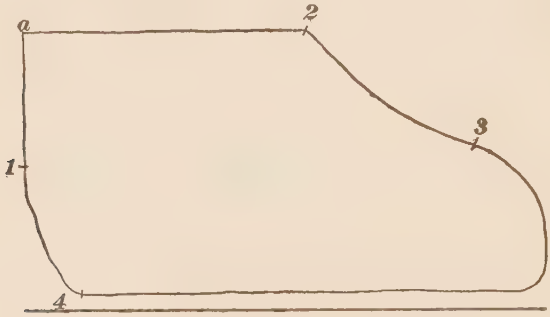


FIG. 21.

When this is the case, a great deal of the benefit of expansion is lost and there is a consequent waste of steam. With an ordinary plain slide-valve engine, the cut-off is always late, it not being practicable to cut off earlier than ½ stroke without seriously affecting the other events.

40. Case V. Release Too Early.—The appearance of the diagram for this case is illustrated in Fig. 18.

41. Case VI. Release Too Late.—This is illustrated in Fig. 19.

42. Case VII. Compression Too Early.—Figs. 20 and 22 show the effects of too early compression. The steam is compressed in the clearance space until its pressure rises above that of the steam in the steam chest; when the steam valve opens there is a flow of steam from the cylinder to the steam chest, as is shown by the loop, until the pressures in the cylinder and steam chest are nearly equal. If the steam valve has no lead, the compression line may rise above the admission line, as shown at *F* in Fig. 20; with lead, the loop will have the form shown in Fig. 22 and at *A* in Fig. 20.

Too much compression is likely to produce an effect on the running of the engine similar to that produced by too early admission; pounding and heating are often to be ascribed to this cause. It also reduces the effective work of the steam.

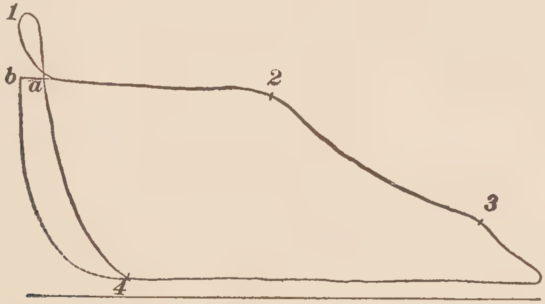


FIG. 22.

With the same cut-off and the proper amount of compression, an amount of work that is represented by the area *a b 4 a*, Fig. 22, included between the line *4 a* and the dotted line *a b 4*, plus the area of the loop, would be gained.

43. Case VIII. Compression Too Late.—This is shown on the diagram by a very short compression curve (see *4–1*, Fig. 19) or by the almost complete absence of such a curve.

REMEDIES FOR FAULTS IN STEAM DISTRIBUTION.

44. Most of the faults enumerated in the preceding articles are due either to incorrect valve proportions or to a fault in the setting; the remedy to be applied in any particular case will, therefore, be determined by a careful consideration of the type of engine and the conditions under which the diagram was taken. With a plain slide valve driven by a fixed eccentric, a change in the angle of advance of the eccentric will have an effect on all the events. Increasing the angle of advance will make admission, cut-off, and release take place earlier and increase the compression; decreasing the angle will have the opposite effect. A general rule is that an increase in the angle of advance of any eccentric with a fixed throw has the effect of making all the events controlled by that eccentric take place earlier, while a decrease in the angle will make them take place later. This rule applies to all types of valves and gears.

It is generally desirable to set the valves so that the work done will be equally divided between the two ends of the cylinder. If an indicator diagram shows a material difference in the work done in the two ends of a cylinder with a slide valve, the fault can be remedied by changing the length of the valve stem so as to make the cut-off take place earlier on the end doing the greater amount of work and later on the other. With an engine having a separate steam valve for each end of the cylinder, either valve may be adjusted so as to make it cut off earlier or later without affecting the other.

45. Remedy for Too Early Cut-Off.—The fault illustrated in Case III and Fig. 20 cannot be remedied by a change in the valve. It is found only in the case of automatic or adjustable cut-off engines working with a light load and a high boiler pressure. The cause is a steam pressure too great for the work to be done with the given size of cylinder and piston speed. The remedy is either to throttle the steam, lower the boiler pressure, or run the engine at a slower speed. That any of these remedies will have the desired effect will be made clear by a consideration of Fig. 23.

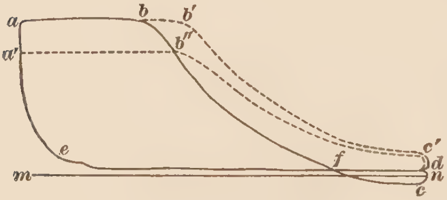


FIG. 23.

The solid line *a b c* represents the expansion line of a diagram when cut-off is so early that the steam expands below the back-pressure line *d e*. The work done is represented by the area *a b f e* minus the area *f d c*. Now let the steam be throttled or the boiler pressure be lowered so that the initial pressure will rise only to *a′*. In order that the work done in the cylinder may be the same with this pressure as it was with the higher pressure, it is evident that cut-off must take place enough later to raise the expansion line $b'' c'$ far enough above the expansion line $b\ c$ to give the diagram $a'\ b''\ c'\ d\ e\ a'$ an area equal to the net area represented by the full-line diagram. On account of the later cut-off, the terminal pressure does not fall below the back-pressure line and no loop is formed.

If the number of revolutions per minute is reduced, the total work done by the engine remaining constant, it is evident that the work done during each stroke must be increased; this will require the admission of steam during a greater portion of the stroke, so as to produce a diagram having a greater area; expansion will begin later and the expansion line will be prevented from falling below the back-pressure line, as is indicated by the dotted line $b'\ c'$.

46. Remedy for Too Late Cut-Off.—With an *adjustable cut-off*, the fault illustrated by Fig. 21 can be remedied either by raising the boiler pressure so that the same area of diagram will be obtained with an earlier cut-off or by increasing the number of revolutions per minute so as to do the same work with a smaller average pressure. By either method the cut-off will be made to take place earlier in the stroke, and the expansion line will, in consequence, fall nearer to the back-pressure line.

47. Release Too Early or Too Late.—If the release is too early, there is danger of loss of pressure due to the escape of the steam too early in the stroke; on the other hand, if the release is too late, the escape of the steam will be so much retarded that the back pressure at the beginning of the return stroke will be excessive. Either of these will represent a loss of work. The valve should be so designed and set that the drop from the expansion line to the back-pressure line will occur as nearly as possible at the end of the stroke. If the engine is provided with separate steam and exhaust valves, this condition will best be fulfilled by setting the exhaust valves so that one-half of the fall in pressure occurs before the piston begins its return stroke;

the release line will then have a form that is well shown in Fig. 15. With a single-valve engine it is often very difficult to secure a satisfactory release line without seriously affecting the other events controlled by the valve.

48. Rules for Compression.—The best indication that the amount of compression is satisfactory is a quiet- and cool-running engine. At the end of the stroke the reciprocating parts must have their direction of motion changed; a force must act to stop them and reverse their direction of motion. By closing the exhaust before the end of the stroke, a part of the steam that would otherwise escape from the exhaust pipe and be lost is retained in the cylinder and acts as a cushion that helps to stop the reciprocating parts quietly. The energy given up by the reciprocating parts in being brought to rest, instead of being wasted in the production of knocks that would result in heating and rapid wear in the bearings, is stored in the steam compressed in the clearance space. The clearance space is thus filled with steam at a pressure more nearly equal to the boiler pressure, and the quantity of steam that must be taken from the boiler is correspondingly reduced.

If there is too little compression, the reciprocating parts will not be satisfactorily cushioned; if there is too much compression, the energy due to the motion will be absorbed before the end of the stroke; the piston must then be pushed by the crank. In either case the effect will be a sudden reversal in the pressures on the bearings that will produce shocks and heating.

49. No simple rule for determining the exact amount of compression to use can be given; however, it may be stated that the amount of compression required to secure quiet running varies with the speed of the engine, but in no case should the compression line extend above the initial or boiler pressure.

It is average practice to compress to about $\frac{9}{10}$ the initial pressure with high-speed engines, $\frac{5}{10}$ with medium-speed engines, and from $\frac{2}{10}$ to $\frac{3}{10}$ with slow-speed engines.

OTHER FAULTS REVEALED BY DIAGRAMS.

50. Introduction.—In addition to faults in valve design and setting, indicator diagrams furnish useful indications of the condition of the piston, cylinder, and valves; insufficient port opening, a steam pipe too small for its purpose, a cramped exhaust passage, or an obstructed exhaust pipe also have an effect on the appearance of the diagram that will generally make it comparatively easy to locate the fault. For a location of some of these faults, it is necessary to draw in the theoretical expansion line, or equilateral hyperbola.

51. Constructing the Theoretical Expansion Line. The general method of using the equilateral hyperbola for testing the expansion line of an actual indicator diagram is illustrated in Fig. 24, where the diagram *E C K L* and the atmospheric line *M N* represent the lines drawn by the indicator. To draw the theoretical expansion line on this diagram, first draw the vacuum line *O X*, as explained in Art. **31.** Perpendicular to *M N* and *O X* draw the two lines *A L* and *B K*, just touching the two ends of the diagram. Measure the length *A B* between these two perpendiculars, and this will give the length of the diagram. Multiply the length so found by the clearance volume of the

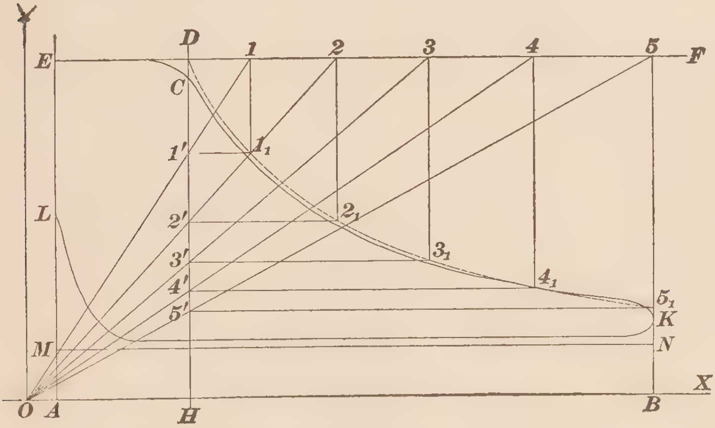


FIG. 24.

end of the cylinder from which the diagram was taken, expressed in per cent., and divide by 100; lay off from *A* a distance *A O* equal to the quotient. From *O* draw the perpendicular *O Y*; this is the **clearance line.** Through the highest point *E* of the steam line draw the horizontal line *E F*. Locate, as nearly as may be done by inspection, the point of cut-off *C*, and through this point draw the perpendicular *D H*. The point *O* where the vacuum line *O X* (the line of no pressure) and the clearance line *O Y* (the line of no volume) intersect represents the point of no pressure and no volume; the distance *A E* or *H D* represents the original absolute pressure; and *O H* represents the original volume of the steam. To obtain points on the theoretical expansion curve, draw lines as *O 1*, *O 2*, *O 3*, *O 4*, *O 5* at random from *O* to the line *E F*. From the points of intersection of these random lines with the line *D H*, as the points $1'$, $2'$, $3'$, $4'$, and $5'$, draw lines parallel to the atmospheric line *M N*. Then, from the points of intersection *1*, *2*, *3*, *4*, and *5* of the random lines drawn from *O* with the line *E F*, drop perpendiculars intersecting the lines drawn from $1'$, $2'$, $3'$, $4'$, and $5'$ in 1_1, 2_1, 3_1, 4_1, and 5_1. These intersections are points on the theoretical expansion line; consequently, through them, by means of an irregular curve, trace the line D–1_1–2_1–3_1–4_1–5_1.

52. Relation Between the Theoretical and the Actual Expansion Lines.

—Numerous tests have shown that when the piston and valves are tight so as to prevent leakage of steam to or from the cylinder after cut-off takes place, the actual expansion line will agree very closely with the theoretical. It will generally be found that the actual line falls somewhat lower than the theoretical, especially towards the end of the stroke, and then raises above the theoretical expansion line, as is shown in Fig. 24 by the crossing of the two lines near 4_1. This is thought to be due to the phenomena known as **cylinder condensation** and **reevaporation,** which may be explained as follows: During the period of exhaust, the cylinder walls are cooled by contact with the relatively cool low-pressure exhaust steam. When the hot steam from the boiler is admitted to the cylinder, a part of it condenses and gives up its latent heat to the cold walls and thus heats them again nearly to the temperature corresponding to the initial pressure; the water formed by this process of condensation is deposited in a thin film on the walls of the cylinder. After cut-off takes place and expansion begins, the pressure of the steam in the cylinder falls until its corresponding temperature is lower than the temperature of the cylinder walls; the walls then give up some of their heat and reevaporate some of the water. The steam thus formed towards the end of the stroke prevents the pressure from falling as fast as it otherwise would and has the effect of raising the actual expansion line somewhat above the theoretical.

53. Leaks Indicated by Expansion Line.—If the actual expansion line departs considerably from the theoretical, it is to be inferred that steam either enters or leaves the cylinder after cut-off takes place. Thus, in Fig. 25, where the expansion line rises very markedly above the theoretical curve, it is evident that the valve leaks and allows steam to enter after cut-off. Similarly, if the expansion line fell below the theoretical curve, the inference would be that steam was escaping from the cylinder through a leaky exhaust valve or past an imperfectly fitting piston. An expansion line that closely follows the theoretical curve is not, however, conclusive evidence that the valves and piston are tight; steam may leak into and out of

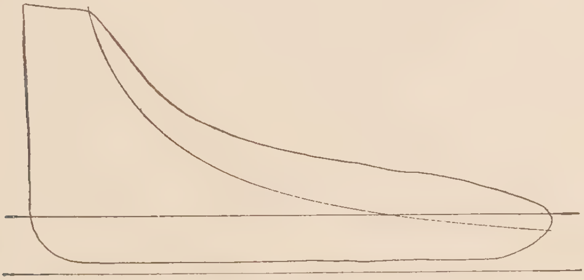


FIG. 25.

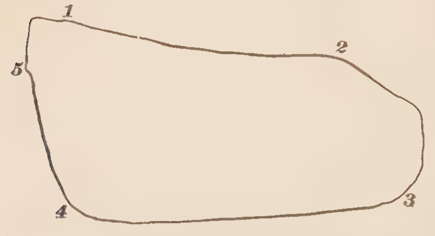
the cylinder at the same time and at such rates that the expansion line will appear to be quite satisfactory.

54. Determining the Point of Cut-Off.—It is sometimes very difficult to determine exactly the point of cut-off from the indicator diagram, especially when the engine has a high rotative speed. The most general method of determining it is to prolong the expansion line upwards by means of an irregular curve and note where it leaves the actual line of the diagram; then take the point of cut-off at or very near the point of deviation (see Figs. 24 and 25).

Instead of locating the point *D*, Fig. 24, at which to begin the theoretical curve by the method just explained, a method sometimes recommended is to prolong the expansion line by means of an irregular curve until it intersects the horizontal through the point representing the initial pressure. The point of intersection is then taken as the point through which to draw the vertical line *D H*, Fig. 24, to represent the volume at point of cut-off.

The rounding of the diagram near the point of cut-off is caused by the slowness of the valve movement at cut-off. On the Corliss and other releasing-gear engines, the valve cuts off very suddenly, the rounding is very slight, and the point of cut-off is very easily located.

55. Leaks Indicated by Compression Line.—If the piston and valves are tight, the compression line will generally curve quite regularly in one direction until it meets the admission line, as is shown by the diagrams in Figs. 15, 16, and 17. It often happens that, as shown at *5* in Fig. 26, the curvature of the compression line changes as the piston nears the end of the stroke; this change sometimes becomes so pronounced as to form a



Scale of Spring=50.

FIG. 26.

hook shown at *a*, Fig. 27. A compression line of this form indicates that steam is escaping from the compression space; the loss may generally be ascribed to a leak either through the exhaust valve or around the piston.

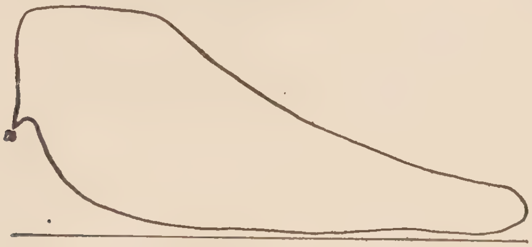


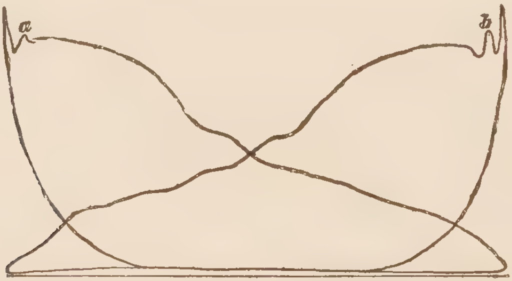
FIG. 27.

56. Effect of Throttling.—When the steam pipe and its connections are of ample size and the steam ports are well opened, a nearly horizontal steam line whose height above the atmospheric line represents a pressure nearly equal to the boiler pressure may be expected, as is shown in Figs. 13 to 16. Any restriction in the passage leading from the boiler to the cylinder has the effect of preventing the flow of steam fast enough to keep the cylinder filled at boiler pressure up to the point of cut-off. This effect is shown on the diagram by a steam line that gradually falls as the piston advances. Fig. 26 is a diagram from an engine with a throttling governor; the effect of the governor in checking the flow of steam to the cylinder is shown by the drop in the steam line between the point *1* and the point *2*, where cut-off takes place. A long steam pipe or a pipe that is too small for its purpose, a partly closed throttle valve, or any similar obstruction will produce a drop in the steam line similar to the one shown in Fig. 26.

A high rotative speed generally results in a drop in the steam line, as is shown by the diagrams in Fig. 17. With shaft-governor engines, especially, the valve opening is often restricted and steam cannot follow the piston fast enough to keep the pressure up to that at the beginning of the stroke.

57. A high back pressure is caused by some obstruction that prevents the free escape of the exhaust. The exhaust from the engine from which the diagram shown in Fig. 26 was taken was discharged into a system of pipes for heating the building, and considerable pressure, as is shown by the height of the back-pressure line *3–4* above the atmospheric line *m n*, is required to force the steam through the coils of pipe. Somewhat similar results will be produced by a choked exhaust port or an exhaust pipe that is very long or too small.

58. Wavy lines on a diagram are generally due to vibrations of the pencil motion when there is a sudden change in pressure, such as takes place when steam is admitted to the cylinder of a high-speed engine or during the period of expansion. The indicator piston and the pencil motion are quickly set in motion and their inertia carries them beyond the point they would reach if the pressure were gradually applied. This effect is well illustrated by the wavy lines at *a* and *b*, Fig. 28, which were produced by vibrations set up by the action of the steam at admission. The expansion lines of the same diagrams show a similar, but less violent, vibration. These effects are common with diagrams from quick rotative speed engines. They are an indication that the indicator piston is in good condition and working freely.



50 Spring, 210 R.P.M., 12"x18" Engine.

FIG. 28.

59. Sticking of the indicator piston is suggested by an expansion line that drops by a series of steps somewhat resembling notches (see Fig. 29). These steps or notches can generally be distinguished from the wavy lines produced by the inertia of the parts of an indicator with a free working piston by their angular appearance.

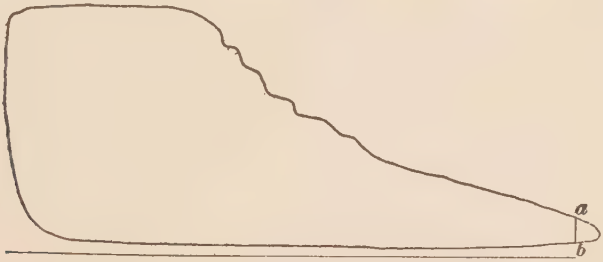


FIG. 29.

60. Striking of the paper drum against the stops is readily detected by the appearance of the release end of the diagram, which, instead of being rounded, as shown in Fig. 27, will have angular corners with a nearly vertical line connecting them, as shown in Fig. 29, where the full line *a b* shows the appearance of the end of the diagram when the drum struck the stop; the dotted line shows the change effected by giving the drum its full range of motion.

ENGINE TESTING.

THERMODYNAMICS OF THE STEAM ENGINE.

INTRODUCTION.

1. Thermodynamics is that branch of physical science that treats of the relation between heat and mechanical work.

2. A **heat engine** is a device by means of which energy, in the form of heat, developed by the combustion of fuel or derived from any other source is transformed into mechanical motion in such a manner that it can be made to do useful work.

3. The Steam Engine a Heat Engine.—The steam engine is in reality a heat engine, steam being merely a vehicle by means of which the heat energy developed by the combustion of the fuel in the furnace is transferred to the moving parts of the engine. From these moving parts the energy is transmitted by such vehicles as shafting, pulleys, belts, and electric currents to the point where it can be made do the required work.

The general principle on which the action of nearly all heat engines is based is the production of an expansive gas or vapor in a confined space. With the steam engine, water is heated in a closed boiler and changed to an expansive

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vapor—steam—whose pressure depends on the temperature to which it is heated. Other easily vaporized liquids, for example, naphtha, are sometimes used instead of water, and the action of their vapors depends on exactly the same principles as govern the action of steam.

4. Work Done by Expansive Force of a Gas.—All gases possess the property of expansibility, by virtue of which they expand and fill the space in which they are confined, no matter how great that space may be. This tendency to expand causes the gas to exert a pressure, called the **tension** of the gas, on the walls of the confining vessel. Keeping this principle in mind, let us consider a given volume of gas confined in a cylinder fitted with a movable piston. The gas in the cylinder tends to expand and thus exerts a pressure on the piston. If the force that resists motion is less than this pressure, the piston will be pushed outwards against the opposing force; the expansive force of the gas overcomes a resisting force, and in so doing does work.

5. Heat the Source of Work Done During Expansion. Careful experiments have shown that there is a fixed relation between work and heat and that heat can be changed into work and work into heat. A study of the effect on the gas of its expansion in the cylinder under such conditions that it does work will show that the work has really been done by heat.

To show that heat is the force that moves the piston, let the cylinder be so made that no heat can get to the gas as it expands; under these conditions a thermometer in the gas would show that it gets colder as it expands and pushes the piston along. The work has been done at the expense of a part of the heat of the gas and its temperature falls. In accordance with the theory of heat, the fall in temperature means that the molecules of the gas move slower; part of the kinetic energy represented by their rate of motion at the beginning of expansion has been expended in doing the work of pushing the piston against the resisting force.

6. If the cylinder is so arranged that enough heat can be added to the gas during expansion to keep its temperature *constant*, and a careful measurement of the heat added and the work done is made, it is found that the quantity of heat added is exactly equal to the heat represented by the work. For example, if the piston is pushed 4 feet by the expanding gas against an average resistance of 5,000 pounds, the work done is 4 × 5,000 = 20,000 foot-pounds. Since 778 foot-pounds of work is equivalent to 1 B. T. U. of heat, it follows that 20,000 ÷ 778, or 25.707 B. T. U. of heat must be added to the gas to make up for the heat expended in pushing the piston and to keep the temperature constant.

7. Adiabatic Expansion.—When a gas expands and does work at the expense of its own heat—no heat being added to it from an outside source—the *expansion* is said to be **adiabatic.**

8. Isothermal Expansion.—When heat is added to a gas so as to keep the temperature constant during expansion, the *expansion* is said to be **isothermal.**

9. Compression of a Gas.—If we have a quantity of gas in a cylinder and push the piston inwards, so as to compress the gas and make it occupy a smaller space, we must do work in overcoming the expansive force of the gas; this work represents a certain amount of energy that is transferred to the gas. If the compression takes place under such conditions that no heat can leave the gas during the change in its volume, the energy represented by the work done on it will appear as heat and the temperature of the gas will be raised; under these conditions we have **adiabatic compression.**

If the compression takes place under such conditions that the heat represented by the work done is removed from the gas so as to keep its temperature constant, the *compression* is **isothermal.**

10. Relation Between Expansion and Compression. The quantity of work that must be done in compressing a gas adiabatically or isothermally from a given volume to a smaller one is exactly equal to the work that the gas can do when expanding, in the same way in which it was compressed, from the smaller volume to the original. Also, the rise in temperature during adiabatic compression and the quantity of heat that must be abstracted when the compression is isothermal are, respectively, equal to the corresponding fall of temperature and the quantity of heat that must be added during adiabatic and isothermal expansion.

11. Relation Between Work and Heat During Expansion or Compression.—In practice, it is seldom that the expansion is purely adiabatic or isothermal. No cylinder can be so made as to absolutely prevent the transfer of some heat to or from the gas, and it is difficult to impart or abstract heat so as to keep the temperature uniform. In any case, however, it is always found that there is a definite relation between the work done and the sum of the quantities of heat represented by the change in temperature of the gas and the heat imparted to or abstracted from it. This relation shows conclusively that the work done by an expanding gas is always a change of heat to work.

12. Expansion Diagrams.—The relation between the pressure and the volume of a gas during expansion may be represented by means of a graphical diagram. To illustrate, consider a cylinder *A*, Fig. 1, in which a piston *P* fits. The cylinder is attached to a reservoir *R* by a pipe *T* that permits air from *R* to enter the space *S* when the valve *V* is opened. A gauge *G* graduated so as to indicate absolute pressures, that is, so that the pointer stands at zero when there is a perfect vacuum in the space *S*, shows the pressure in the cylinder; a cock *C*, when opened, permits any air in the cylinder to escape when the piston is pushed back. Now, with the valve *C* open, push the piston clear back to the end of the cylinder, thus forcing out all the air; then close *C* and open *V*, so as to admit air from *R*, in which there is a constant pressure. Permit the piston to move slowly to the left with *V* open, and the gauge shows a constant pressure of the air in the space *S*. When the piston has moved a certain distance to the left, close *V*, so as to stop the admission

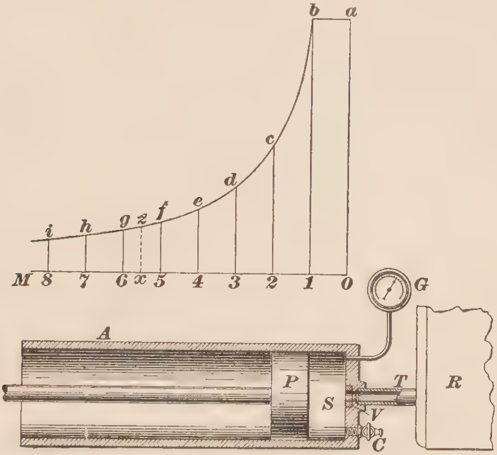

FIG. 1.

of air from the reservoir. Now, as the piston is permitted to move farther to the left, the gauge shows that the pressure falls. If the temperature in the cylinder is kept constant, it is found that when the piston is twice its original distance from the end and the air has expanded to twice its original volume, the pressure, in accordance with Mariotte's law, is only one-half the original pressure. When the volume is three times as great as the original volume, the pressure is found to be one-third the original pressure. When the volume has increased four times, the pressure is one-fourth as great, etc.

13. To represent this action graphically, draw a line *O M* to represent the piston motion and divide this line into a number of equal parts, *O–1*, *1–2*, *2–3*, etc., each of which, to some convenient scale, represents a motion of the piston through a distance equal to that through which it moved while *V* was open. Since the volume of air in the cylinder is proportional to the distance of the piston from the end of the cylinder, each of the sections *O–1*, *O–2*, etc. represents a volume equal to the original volume of air admitted to the cylinder from the reservoir, and the distances *O–1*, *O–2*, *O–3*, etc. represent the volume of the air in the cylinder for piston positions corresponding to the points *1*, *2*, *3*, etc.

From *O* draw a vertical line *O a*, and, to some convenient scale, make its length represent the pressure at the beginning of the piston stroke. Draw other vertical lines from the points *1*, *2*, *3*, etc., and, to the same scale as that to which *O a* was drawn, make their lengths represent the pressures corresponding to the piston positions represented by the points *1*, *2*, *3*, etc. and to the volumes represented by the distances *O–1*, *O–2*, *O–3*, etc. Since the pressure, when the piston is at *1*, is the same as the pressure at the beginning of the stroke, the length of the perpendicular *1–b* is the same as the length of *O a*. At *2* the volume is *O–2*, twice the original volume, and if the expansion is isothermal, the pressure is one-half the pressure at *1*; consequently, the length of the line *2–c* is one-half the length of *O a* or *1–b*.

Any desired number of points *c*, *d*, *e*, *f*, etc. can be located and a curve drawn through them. The distance of any point *x*, on the line *O M*, from the point *O* represents, to the scale of volumes, the volume of air in the cylinder when the piston is in the position corresponding to this point; likewise, the vertical distance *x z* from the point *x* to the curve represents, to the scale to which the pressures were laid off, the pressure for the corresponding piston position and volume.

14. The Isothermal Expansion Line, or Equilateral Hyperbola.—The curve that represents the relation between the pressure and volume when the temperature is constant is called the **isothermal expansion line.** This curve follows the law of the curve known in mathematics as the **equilateral hyperbola**; it is, therefore, often called by that name.

15. The Adiabatic Expansion Line.—If no heat is added to the air as it expands, that is, if the expansion is adiabatic, the gauge *G*, Fig. 1, shows a more rapid drop in pressure as the piston advances; each vertical line representing the pressure in the cylinder after expansion begins is shorter than the corresponding line of the isothermal curve; the curve drawn through their upper ends will, therefore, fall below the isothermal curve. The curve representing adiabatic expansion is called the **adiabatic expansion line.**

16. Comparison of the Isothermal and Adiabatic Expansion and Compression Lines.—In Fig. 2, the

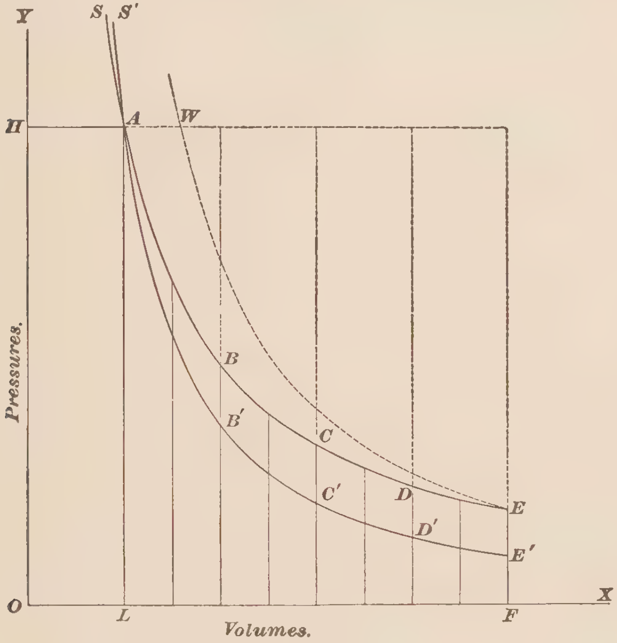

FIG. 2.

curve $A\,B\,C\,D\,E$ represents the isothermal, and $A\,B'\,C'\,D'\,E'$ the adiabatic, expansion line of a quantity of air whose original volume and pressure are, respectively, represented by the distances $O\,L$ and $O\,H$ or $L\,A$. If the air were compressed isothermally from the volume $O\,F$ and the pressure $F\,E$, the pressure would rise as the volume decreased, and the curve $E\,D\,C\,B\,A$ would show the relation between the volume and the pressure. If, with the same original volume and pressure, the air were compressed adiabatically, the curve representing the relation between the volume and the pressure would rise above the isothermal compression curve, as is shown by the dotted line *E W*. If a quantity of air, whose volume is represented by *O F* and whose pressure by *F E'*, is compressed adiabatically, the curve representing the relation between the volumes and the pressures during the process of compression will be *E' D' C' B' A*, which is the same curve that represented the relation for adiabatic expansion from the volume *O L* and the pressure *L A*.

17. Expansion of Steam.—When steam expands and does work, there is the same relation between heat given up and work done as has been explained for gas. Owing, however, to the properties of saturated steam, by virtue of which the pressure depends solely on the temperature and is independent of the volume, the relation between volume and pressure is not as simple as is the case with a perfect gas. For example, if a given weight of dry saturated steam expands adiabatically, a part of it will be condensed; while if the expansion is isothermal, the steam will be superheated during its expansion. If there is a mixture of steam and water, that is, if there is water in the vessel in which the steam expands, the relation between volume and pressure during expansion depends on the proportion of water in the mixture. As long as there is water present, the steam will be saturated and the pressure during isothermal expansion will be constant. This will be evident if we consider the fact that the pressure of saturated steam (steam in contact with water) depends solely on the temperature; if the temperature is constant, the pressure must also be constant, no matter what the volume may be. During isothermal expansion, the heat that is added merely changes some of the water to vapor, which fills the increased space, and there is no change in the pressure of the original steam.

If water is present during the adiabatic expansion of steam, it will give up some of its heat to assist the steam in doing its work; in consequence of the heat derived from the water, the temperature and pressure of the steam will fall slower during expansion as the quantity of water from which heat can be derived is greater.

18. Expansion Curve of Steam.—A consideration of the above outline of the effect of water on the expansive action of steam will make it clear that an innumerable variety of curves, depending on the quantity of water present and the conditions under which expansion takes place, will correctly represent the relation between the pressures and volumes for the expansion of saturated steam. It has, however, been found that under the conditions generally existing in the cylinder of a steam engine, the curve that most nearly represents the relation between pressures and volumes is the **equilateral hyperbola,** which is the curve that shows the relation between the pressures and the volumes of a perfect gas when it expands according to Mariotte's law.

CALCULATING THE WORK DONE ON A MOVING PISTON.

19. Net or Effective Pressure.—A piston that is being pushed through a cylinder by the expansive force of a gas or vapor acting on one side must generally overcome the resisting force of a gas or vapor on its opposite side. Thus, in Fig. 3, let the space in the cylinder at the left of the piston be in communication with the steam space of a boiler in which there is an absolute pressure of 100 pounds per square inch, while the space at the right is open to the atmosphere and, in consequence, is filled with vapor at a pressure of about 14.7 pounds

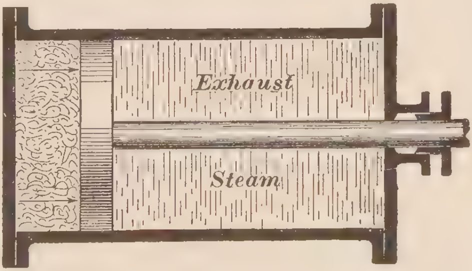

FIG. 3.

per square inch. It is evident that, neglecting the friction of the piston in the cylinder, the force that can be transmitted through the piston rod, and so made do work in overcoming some outside resistance, is the difference between the total pressure of the steam on the left and the total pressure of the air on the right of the piston. This difference is called the **net** or **effective pressure** on the piston. Since the pressure of the atmosphere is 14.7 pounds per square inch, and if the area of the piston is 100 square inches and the absolute pressure of the steam 100 pounds per square inch, the net pressure on the piston in Fig. 3 is $(100 - 14.7) \times 100 = 8{,}530$ pounds.

20. Rule for Calculating Work When Net Pressure and Piston Displacement Are Known.—The work done as the piston moves from one end of the cylinder to the other may be found as follows:

Let $P =$ the net pressure per square foot exerted on the piston;
$A =$ area of piston in square feet;
$L =$ distance in feet moved over by the piston.

Then, the total net pressure on the piston is $P \times A$ pounds, and the distance through which this pressure acts is L feet. The work done is the force multiplied by the distance, or $PA \times L = PAL$ foot-pounds. But AL equals the area of the piston multiplied by the length of the stroke, which equals the volume displaced by the piston during its movement from one end of the cylinder to the other. Let V represent this volume expressed in cubic feet. Then, letting W represent the work in foot-pounds, we have $W = PAL = PV$.

It is usually more convenient to express pressures in pounds per square inch instead of pounds per square foot. Let p represent the net pressure on the piston in pounds per square inch.

Then, $$P = 144p,$$

and $$W = PV = 144pV.$$

Rule 1.—*To find the work done by a piston moving in a cylinder, multiply 144 by the net pressure on the piston in pounds per square inch and by the volume displaced by the piston expressed in cubic feet. The result will be the work in foot-pounds.*

The same result will be obtained by multiplying the pressure in pounds per square inch by the volume displaced by the piston in cubic inches and dividing the result by 12.

The volume displaced by a piston during a single stroke or a given period of time is often called the piston displacement for the stroke or the given period.

EXAMPLE.—The piston of an engine is acted upon by a net pressure of $32\frac{1}{2}$ pounds per square inch. The volume swept through by the piston at each stroke is $5\frac{1}{2}$ cubic feet. (*a*) How much work is done at each stroke? (*b*) If the engine makes 80 strokes per minute, what horsepower does it develop?

SOLUTION.— (*a*) According to rule **1**, the work

$$W = 144 \times 32\tfrac{1}{2} \times 5\tfrac{1}{2} = 25{,}740 \text{ ft.-lb.} \quad \text{Ans.}$$

(*b*) The number of foot-pounds per minute is $25{,}740 \times 80$, and the horsepower developed is, therefore,

$$\frac{25{,}740 \times 80}{33{,}000} = 62.4 \text{ H. P.} \quad \text{Ans.}$$

21. Work Diagrams.—The work done by a moving piston may be represented by a diagram similar to the diagrams used to represent the relation between the volumes and pressures of an expanding gas or vapor. For example, in Fig. 4, two lines OX and OY are drawn at right angles, the line OX being horizontal and the line OY vertical. Suppose that the area of the piston is 2 square feet and that the distance moved by it is 6 feet. Then, when the piston moves 1 foot, it displaces a volume of 2 cubic feet. On the line OX lay off a distance O–*1*, and let this distance represent a piston travel of 2 feet. Then, the distance O–*2*, which is twice O–*1*, represents a piston travel of 2×2 feet $= 4$ feet, and, similarly, the distance O–*3*, which is three times the distance O–*1*, represents a travel of 3×2 feet $= 6$ feet.

Since the piston area does not change, the volume swept through is proportional to the piston travel; therefore, *O–1* may be taken to represent the displacement when the piston has traveled 2 feet. That is, *O–1* represents a volume of 2 × 2 = 4 cubic feet, *O–2* represents 8 cubic feet, and *O–3*, 12 cubic feet. The piston is supposed to be moving from left to right, that is, in the direction *O X*.

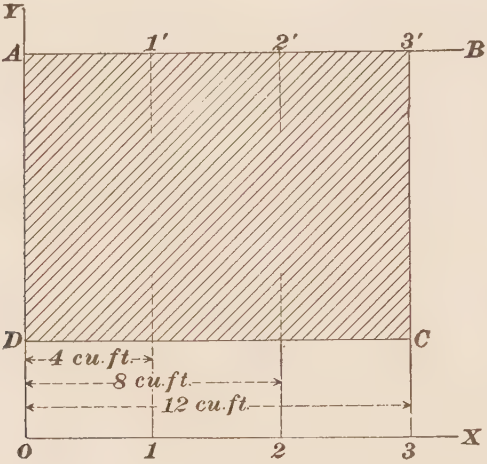

FIG. 4.

When the piston is at the beginning of its travel, that is, at the position represented by *O Y*, lay off on the line *O Y* a distance *O A*, which, to the scale selected, represents the pressure on the left side of the piston. Suppose the pressure is 60 pounds per square inch. Then, if *O A* is 2 inches, the scale is $\frac{60}{2} = 30$ pounds; that is, a vertical height of 1 inch represents 30 pounds per square inch pressure. Suppose the pressure to be the same throughout the stroke. Then, when the piston is at the point represented by *1*, the pressure is represented by the distance *1–1′*, which is equal to *O A*. Likewise, when the piston is in positions *2* and *3*, the distances *2–2′* and *3–3′*, respectively, represent the pressures at those points. In brief, the pressure upon the left side of the piston at any position may be found by measuring the vertical distance between the lines $O\,X$ and $A\,B$ at that point and multiplying by the scale, 30 pounds per inch of height. In a similar manner, lay off on the line $O\,Y$ a distance $O\,D$, which, to the scale already used, represents the pressure of the atmosphere on the right of the piston, and is, therefore, equal to $\frac{14.7}{30} = .49$ inch. Since this pressure on the right of the cylinder is constant throughout the stroke, the distance from any point on the line $O\,X$ to the line $D\,C$ parallel to $O\,X$ represents the opposing pressure on the piston when it is at the corresponding point of its stroke.

22. The net pressure on the piston is represented by the distance $D\,A$ $(= O\,A - O\,D)$. We have shown that, to the scale selected, $O\text{–}3 = D\,C$ represents the piston displacement. According to rule **1**, the work done by the piston is proportional to the net pressure multiplied by the volume. Now, on the diagram of Fig. 4, $D\,A$ represents the net pressure and $D\,C$ the volume. But $D\,A \times D\,C =$ area $A\,3'\,C\,D$. Hence, the area $A\,3'\,C\,D$ must, to some scale, represent the work done by the piston.

$A\,O$ is 2 inches and $O\,D$ .49 inch; hence the distance $D\,A = O\,A - O\,D$ is $2 - .49 = 1.51$ inches; $D\,C$ equals 2 inches. Therefore, the area of the diagram is $1.51 \times 2 = 3.02$ square inches. The scale of pressure adopted was 1 inch equals 30 pounds per square inch. Hence, $p = 30 \times D\,A$. Since $D\,C$ (= 2 inches) represents 12 cubic feet of volume, the scale of volumes must be $\frac{12}{2} = 6$ cubic feet per inch of length. Hence, $V = 6 \times D\,C$. Then, from rule **1**, the work is

$$
\begin{aligned}
W = 144\,p\,V &= 144 \times (30 \times D\,A) \times (6 \times D\,C) \\
&= 144 \times 30 \times 6 \times (D\,A \times D\,C) \\
&= 144 \times 30 \times 6 \times 3.02 = 78{,}278.4 \text{ foot-pounds.}
\end{aligned}
$$

23. The diagram may be used in another way. The distances $O\text{–}1$, $O\text{–}2$, and $O\text{–}3$ may represent the distances moved through by the piston instead of the volumes displaced by it. Then, $D\,C$ represents the stroke of the piston, in this case 6 feet, and since $D\,C = 2$ inches, the horizontal scale is $\frac{6}{2} = 3$ feet of piston travel = 1 inch of length.

The work is

$$W = 144\,p\,A\,L.$$

As before,

$$p = 30 \times DA,$$
$$L = 3 \times DC,$$

and $$A = 2 \text{ square feet.}$$

Hence, $$\begin{aligned} W &= 144 \times (30 \times DA) \times 2 \times (3 \times DC), \\ &= 144 \times 30 \times 2 \times 3 \times (AD \times DC) \\ &= 78{,}278.4 \text{ foot-pounds.} \end{aligned}$$

The latter method is the one usually employed in calculating the horsepower of an engine by means of the indicator diagram.

24. Diagrams for Varying Pressures.—The diagram of Fig. 4 is very simple, because the pressure on both sides of the piston is constant throughout the stroke, thus making the diagram a rectangle. Suppose the pressure decreases uniformly throughout the stroke, as shown in Fig. 5. Here the net pressure at the beginning of the stroke is represented by the distance DA, and at the end of the stroke by the distance CB. To calculate the work, it is necessary to find the average net pressure throughout the stroke. In this case the diagram is a trapezoid; the average pressure is, therefore, represented by the line $HG = \frac{1}{2}(DA + CB)$. This distance HG is called the **mean ordinate** of the diagram $ABCD$. It has such a length that, being multiplied by the distance DC, it will give the area of a rectangle $EFCD$ that will be equal to the original area $ABCD$. The work is found by multiplying this mean ordinate by the length DC, then by the scales of pressures and volumes, and by 144.

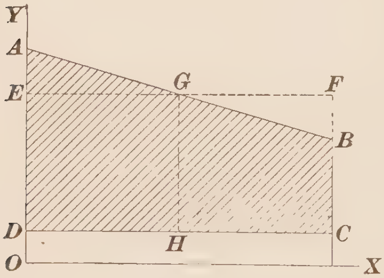

FIG. 5.

25. In Fig. 6 diagrams taken from both sides of the piston of an actual steam engine are shown on the same card. The line *A B* represents the varying steam pressure during the forward stroke and the line *B C D* of the crank-end diagram represents the back pressure opposing the motion of the piston during the forward stroke. Hence, the net pressure at any piston position is given by the vertical distance between the line *A B* of one diagram and the line *D C B* of the other diagram at the point representing the piston position. Likewise, the net pressure for any point of the return stroke is given by the vertical distance between the line *E D* of the crank-end diagram and the back-pressure line *B F D* of the head-end diagram. The net work done by the piston, as in the preceding cases, is given per stroke by the area *A B C D* for the forward stroke and the area *E D F B* for the return stroke. It will be noticed that the area *B C D F* has been taken from one diagram and added to the other diagram.

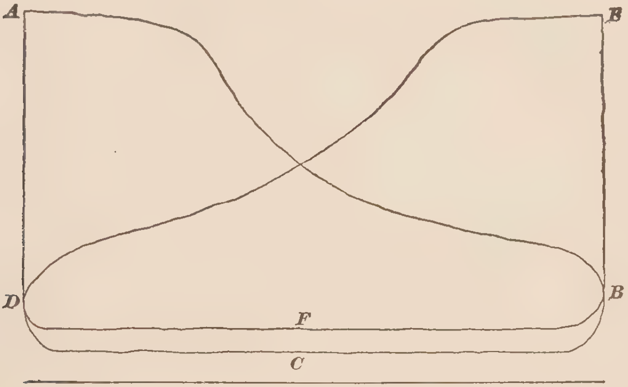

FIG. 6.

Now, to find the average work per stroke of a double-acting engine, the sum of the areas representing the work done during the forward and return stroke is divided by 2. Evidently, the sum of the areas will be the same whether we add the areas *A B C D* and *E D F B* or add the areas of each diagram, as $A\,B\,F\,D$ and $E\,D\,C\,B$. Hence, the *average work* will be correctly given by considering the area of each diagram as representing the work done on the side of the piston the diagram was taken from and dividing the sum of the areas by 2. While the assumption that the area of each diagram represents the net work done on its side of the piston of a double-acting engine is not entirely correct, it is, nevertheless, a very convenient assumption to make, and will *not* cause any error in finding the *average pressure* per stroke when both diagrams are considered. The convenience of making the assumption just explained is best exemplified in case of diagrams taken on separate cards; in that case it would be necessary to very carefully transfer the back-pressure lines from one card to the other in order to get the correct area representing the work done on each side of the piston. This is a tedious operation calling for considerable skill in the use of drawing instruments; the necessity for this operation is obviated by making the assumption stated.

In a single-acting engine, which takes steam on one side of the piston only, the other side of the piston being open to the atmosphere, the area of the diagram represents correctly the work done during the revolution. This can readily be seen by a consideration of Fig. 7, where $O\,E$ is the

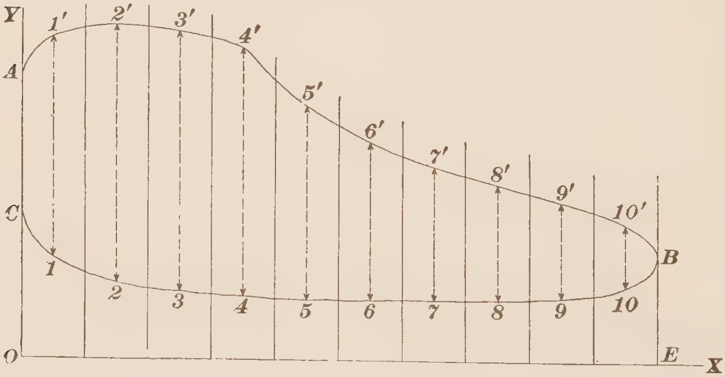

FIG. 7.

atmospheric line. The work done by the piston during the forward stroke is given by the area *O A B E*, and the work that must be done on the return stroke to make the piston return to the beginning of the working stroke is given by the area *O C B E*. Hence, the net work is equal to the difference of the areas *O A B E* and *O C B E*, which is the area *A B C* (the area of the diagram).

26. To find the area of an indicator diagram, we must find its mean ordinate. This may be done approximately in the following manner: Divide the length *O E* of the diagram (see Fig. 7) into a number of equal parts (10 or 20 parts are most convenient) and through each division draw a vertical line. Half way between these vertical lines draw the lines *1–1′*, *2–2′*, *3–3′*, etc., extending between the lines *A B* and *B C*. These vertical distances between the two curves are called **ordinates.** As shown in the figure, there are ten of these ordinates equally distant from each other. If their lengths are all added together and the sum divided by the number of ordinates, the result is the average distance between the lines, or the mean ordinate.

This ordinate multiplied by the distance *O E* gives the area of the diagram. Usually both the ordinate and *O E* will be measured in inches; the area will then be expressed in square inches. The area being found, the work is calculated by rule **1.** That is, multiply the area by the vertical scale of pressures, by the horizontal scale of volumes, and by 144. The result is the work in foot-pounds.

EXAMPLE.—The area of a diagram like that shown in Fig. 7 is found to be 7.34 square inches. The vertical scale of pressure is 36 pounds equals 1 inch, and the horizontal scale of volumes is $2\frac{1}{2}$ cubic feet equals 1 inch. What is the work per stroke of piston ?

SOLUTION.—Multiply the area by the horizontal and vertical scales, and by 144, or work $= 7.34 \times 36 \times 2\frac{1}{2} \times 144 = 95{,}126.4$ ft.-lb. Ans.

27. Work Diagram for Expanding Steam.—In connection with the diagram of Fig. 4, the piston area was taken as 2 square feet and the length of stroke as 6 feet. Fig. 8 shows the pressure diagram on the supposition that steam from the boiler is shut off when the piston has reached one-third its stroke. Up to that point steam has entered from the boiler at a constant pressure, shown by the line *A B*. The volume of steam in the cylinder at this point is 4 cubic feet. As the piston moves forwards, the pressure begins to fall. When two-thirds the stroke is completed, the steam that previously occupied 4 cubic feet now occupies 8 cubic feet, that is, its volume is doubled, and assuming the expansion to follow Mariotte's law, its pressure should be one-half what it was originally, or $\frac{1}{2}$ *b B*. When the piston reaches the end of the stroke, the steam occupies 12 cubic feet, or three times its original volume. Therefore, its pressure is one-third the original pressure; that is, $e E = \frac{1}{3} b B$. The line *B E* shows the fall of pressure during the last two-thirds of the stroke. When steam is shut off from the boiler in this manner and does work at the expense of its own heat, it is said to be used expansively.

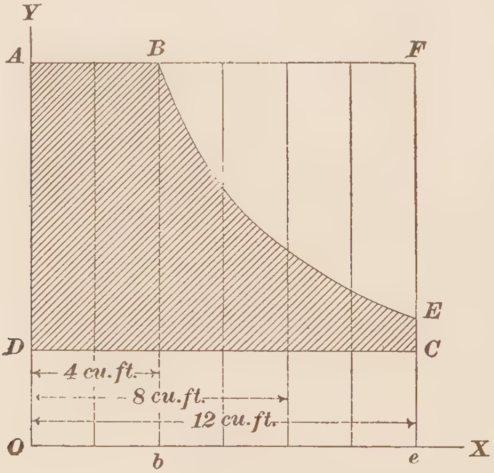

FIG. 8.

28. Gain in Work by Using Steam Expansively.—It was found that the area of the diagram of Fig. 4, which represents the work done when the steam followed the piston full stroke, was 3.02 square inches, and the work done per stroke of piston was 78,278.4 foot-pounds. The area of the diagram *A B E C D*, Fig. 8, which represents the work done when the steam supply to the cylinder was cut off when one-third of the piston stroke was completed, is nearly 1.82 square inches. The work done per stroke is, therefore, $1.82 \times 30 \times 6 \times 144 = 47,174.4$ foot-pounds.

In the first case, a cylinder full of steam, 12 cubic feet, was taken from the boiler, and the work obtained from each cubic foot was, therefore, $\frac{78,278.4}{12} = 6,523.2$ foot-pounds. In the second case only 4 cubic feet of steam was taken from the boiler. Consequently, the work done by each cubic foot of steam used was $\frac{47,174.4}{4} = 11,793.6$ foot-pounds, or nearly twice as much as was done by a cubic foot when the steam followed the piston for the full stroke.

EXAMPLES FOR PRACTICE.

1. The mean ordinate of a diagram similar to that shown in Fig. 7 is 1.2 inches long. The vertical scale of pressure is 1 inch = 40 pounds per square inch, and the horizontal scale of distances is 1 inch = 10 inches. The length of the diagram is 3 inches, and 1 foot of actual length of the vessel that contains the steam represents a volume of 452 cubic inches. What is the work done in one stroke of the piston? Ans. 4,520 ft.-lb.

2. The mean ordinate of a diagram is .89 inch; the length of the diagram, 3.2 inches; the vertical scale of pressures, 1 inch = 50 pounds per square inch; the horizontal scale of volumes, 1 inch (diagram) = .56 cubic foot. Find the work done in 12 strokes.
Ans. 137,797.6 ft.-lb.

HORSEPOWER OF STEAM ENGINES.

INDICATED HORSEPOWER AND NET HORSEPOWER.

29. The relation between the pressures on the two sides of a moving piston and the work done on the piston was explained in Arts. **19** to **26**, and the student is advised to carefully review the explanation there given in conjunction with his study of this section. When the work done in a given period of time is known, the corresponding horsepower is obtained as follows: *Having the work given in foot-pounds per minute, to find the horsepower divide by 33,000; if the work is given in foot-pounds per second, the horsepower is found by dividing by 550.* Horsepower is often abbreviated to H. P.

30. Indicated Horsepower.—The indicator furnishes the most ready method of measuring the pressures on the piston of a steam engine and, in consequence, of determining the amount of work done in the cylinder and the corresponding horsepower. The power measured by the use of the indicator is called the **indicated horsepower.** It is the total power developed by the action of the net pressures of the steam on the two sides of the moving piston. The indicated horsepower is generally represented by the initials I. H. P.

31. Friction horsepower is the part of the indicated horsepower that is absorbed in overcoming the frictional resistances of the moving parts of the engine. If the engine is running light—with no load—all the power developed in the cylinder is absorbed in keeping the engine in motion, and the friction horsepower is equal to the indicated horsepower. This principle furnishes a simple approximate method of finding the friction horsepower of a given engine; since, however, the friction between the surfaces increases with the pressure, the power absorbed in overcoming engine friction will be greater as the load on the engine is increased.

32. Net horsepower is the difference between the indicated and the friction horsepower. It is the power the engine delivers through the flywheel or shaft to the belt or the machine driven by it, and is sometimes called the **delivered horsepower.** Since the power an engine is capable of delivering when working under certain conditions is often measured by a device known as a *Prony brake*, the net horsepower is also called the **brake horsepower.**

33. The **mechanical efficiency** of an engine is the ratio of the *net horsepower* to the *indicated horsepower*; or it is the percentage of the mechanical energy developed in the cylinder that is utilized in doing useful work.

To find the efficiency of an engine, when the indicated and net horsepowers are known:

Rule 2.—*Divide the net horsepower by the indicated horsepower.*

Example.—The indicator diagrams taken from an engine running under full load show the I. H. P. to be 238.5. The diagrams taken when the engine is running at the same speed under no load show a horsepower of 39.7. (*a*) What is the approximate net H. P. developed by the engine? (*b*) What is the efficiency of the engine?

Solution.— (*a*) Approximate net H. P. = I. H. P.—friction H.P. =

$$238.5 - 39.7 = 198.8. \quad \text{Ans.}$$

(*b*) By rule **2**, the efficiency is

$$\frac{198.8}{238.5} = .834 = 83.4 \text{ per cent.} \quad \text{Ans.}$$

The mechanical efficiency of a good engine is from 75 to 90 per cent.

MEASURING THE INDICATED HORSEPOWER.

34. In accordance with the principles explained in Arts. **19** to **26**, when the net pressure on the piston and the piston displacement for a given period of time are known, the work done during the given period can be calculated. The usual period of time considered when calculating the power of an engine is 1 minute; since 33,000 foot-pounds of work per minute is equal to 1 horsepower, the horsepower is obtained by dividing the work done in one minute by 33,000.

FINDING THE M. E. P.

35. The **mean effective pressure**, or M. E. P., is defined as the average pressure urging the piston forwards during its entire stroke in one direction, less the pressure that resists its progress.

36. The mean effective pressure may be found in three ways:

1. The area of the diagram in square inches may be measured by an instrument called the *planimeter;* the M. E. P. is then found by dividing the area of the diagram in square inches by the length of the diagram in inches and multiplying by the scale of the spring.

EXAMPLE.—The area of the diagram is 4.2 square inches and the length is 3.5 inches; a 40 spring being used, find the M. E. P.

SOLUTION.— $\frac{4.2}{3.5} \times 40 = 48$ lb. per sq. in., M. E. P. Ans.

2. A special form of planimeter may be used by means of which the M. E. P. may be measured directly.

3. Where a planimeter is not available, the M. E. P. may be found with a fair degree of accuracy by multiplying the length of the mean ordinate by the scale of the spring.

37. The Planimeter.—A common form of this instrument is shown in Fig. 9. It consists of two arms hinged together by a pivot joint at *j*. One arm carries a recording wheel *d*, which rolls on the surface to which the card is fastened, while the outline of the diagram is being traced by the point *f*. The needle point *p* is fixed in the paper or drawing board, and remains stationary during the operation.

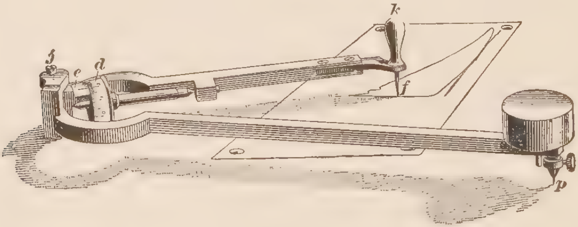

FIG. 9.

The indicator card should be fastened to a smooth table or drawing board that has been previously covered with a piece of heavy unglazed paper or cardboard. The point *p* should be placed far enough from the card to enable the wheel to roll on the unglazed paper without touching the card, as it will slip if rolled over a smooth surface. Set the zero of the wheel *d* opposite the vernier *e*; then, with the tracing point *f*, follow the line of the diagram carefully, *going around the diagram in the direction of the hands of a watch*, and stop exactly at the starting point.

38. Reading the Vernier.—The area is read from the recording wheel and vernier as follows: The circumference of the wheel is divided into 10 equal spaces by long lines that are consecutively numbered from 0 to 9. Each of these spaces represents an area of 1 square inch and is subdivided into 10 equal spaces, each of which represents an area of .1 square inch. Starting with the zero line of the wheel opposite the zero line of the vernier and moving the tracing point once around the diagram, the zero of the vernier will be opposite some point on the wheel; if it happens to be directly opposite one of the division lines on the wheel, that line gives the exact area in tenths of a square inch. The zero of the vernier, however, will probably be between two of the division lines on the wheel, in which case write down the inches and tenths that are to the left of the vernier zero, and from the vernier find the nearest hundredth of a square inch as follows: Find the line of the vernier that is exactly opposite one of the lines on the wheel. The number of *spaces on the vernier* between the vernier zero and this line is the number of hundredths of a square inch to be added to the inches and tenths read from the wheel. An example is presented in Fig. 10, where the 0 of the vernier lies between the lines on the wheel representing 4.7 and 4.8 square inches, respectively, showing that the area is something more than 4.7 square inches. Looking along the vernier it is seen that there are three spaces between the vernier zero and the line of the vernier that coincides with one of the lines on the wheel; this shows that .03 square inch is to be added to the 4.7 square inches read from the wheel, making the area

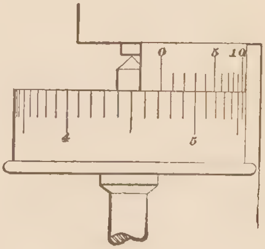

FIG. 10.

4.73 square inches, to the nearest hundredth of a square inch.

39. Modifications of Planimeter.—Planimeters are made in a number of different forms, some of which differ considerably from that shown in Fig. 9. One of the most important modifications in the general form is found in the **Lippincott** and the **Willis** planimeters. In these instruments the wheel, in addition to its rotary motion, slides in the direction of the axis of its spindle, and the area is indicated by the amount of this sliding motion as measured by a scale parallel to the axis. The **Coffin averaging instrument** is another modification, in which the end of the bar to which the wheel and vernier are attached is guided along a straight line by a slot instead of being jointed to another bar.

40. Measuring the M. E. P. Directly.—With the planimeter illustrated in Fig. 9, the M. E. P. is found by dividing the area as measured by the instrument by the length of the diagram and multiplying the quotient by the scale of the spring. Many planimeters, however, including those mentioned in the last article, can be used to measure the M. E. P. directly, no calculation being required. For this purpose, special adjustments and scales are provided by means of which the instrument can be set to correspond to the length of the diagram and the scale of the spring. The makers furnish complete instructions for the use of each of these special attachments.

41. Hints for Use of Planimeter.—It is well to so place the fixed point (*p*, Fig. 9) of the instrument that, as the tracing point moves around the diagram, the arms will swing about equally on each side of a position at right angles with each other. A slight dot is generally made with the tracing point to mark the point at which its motion around the diagram begins; when the tracing point reaches this dot in the paper, the operator knows that the motion around the diagram has been completed. The direction of motion of the tracing point must always be the same as that of the hands of a watch; motion in the opposite direction will move the wheel in the wrong direction and give a negative reading for the area.

When measuring diagrams with loops, like Fig. 11, move the tracing point so that it will follow the outline of the loops in a direction opposite to the direction of motion of the hands of a watch, as is indicated by the arrowheads on the diagram, Fig. 11. This will cause the instrument to automatically subtract the areas of the loops from the area of the main part of the diagram.

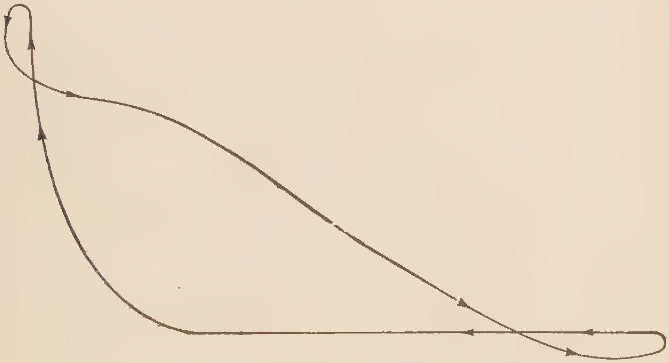

FIG. 11.

An excellent check on the work is to start with the recording wheel at zero and pass the tracing wheel around the diagram two or three times, noting the reading of the wheel each time the tracing point returns to the point of starting. Each reading of the wheel divided by the number of times the outline of the diagram has been traced should give, very nearly, the value of the first reading; if there is a considerable difference between the first reading and the value obtained by dividing the second reading by 2 or the third reading by 3, it is an indication that an error has been made, and the work should be repeated. If the difference is small, the work may be assumed to be satisfactory and the value to be used for the area or the M. E. P. may be taken as the average found by dividing the last reading by the number of times the tracing point passed around the diagram.

42. Finding the M. E. P. by Ordinates.—This operation may be performed by the aid of two triangles, a scale, and a hard lead pencil; if two triangles are not available, a single triangle and a straightedge will suffice. Lines perpendicular to the atmospheric line and tangent to the two ends of the diagram must first be drawn; the perpendicular distance between these tangents will be the length of the diagram, and this length must be divided into some number of equal parts (10 or 20 parts are the most convenient, but any other number may be used). Midway between any of the points of division draw a line parallel to the two tangents; the part of this line included between the lines of the diagram is the middle ordinate of its corresponding space. The sum of the lengths of all of these middle ordinates divided by the number of spaces is the mean ordinate and gives, approximately, the average height of the diagram. The length of the mean ordinate should agree very nearly with the value obtained by dividing the area of the diagram—as measured by a planimeter—by the length of the diagram. The M. E. P. is found by multiplying the length of the mean ordinate by the scale of the spring with which the diagram was taken.

If a scale graduated to correspond with the scale of the spring is available, the M. E. P. may be obtained by measuring the ordinates in pounds instead of in inches; the sum of the lengths of the ordinates as so measured divided by their number gives the M. E. P. of the diagram. For example, let the scale of the spring be 40, then each $\frac{1}{40}$ inch in the length of an ordinate represents a pressure of 1 pound per square inch, and by measuring the length of an ordinate with a scale graduated in fortieths of an inch, the number of pounds pressure represented by that ordinate is found.

43. A convenient method of finding the sum of the lengths of the ordinates of a diagram, and one that is especially to be recommended when a decimal scale is not available, is the following: Take a strip of paper having a straight edge a little longer than the sum of the lengths of the ordinates. Lay this strip along the first ordinate. From the point on the strip representing one end of the first ordinate lay off the length of the next ordinate. In the same way lay off on the strip the length of each of the ordinates in succession. The length of the strip included between the extreme, or first and last, points so marked will be equal to the sum of the lengths of the ordinates, and this length divided by the number of ordinates will give the length of the *mean ordinate.*

EXAMPLE.— (*a*) The lengths between the extreme points on a strip of paper on which has been laid off successively the lengths of the 10 ordinates of an indicator diagram is $12\frac{5}{16}$ inches. What is the length of the mean ordinate to the nearest .001 inch? (*b*) The diagram was taken with a 20 spring; what is the M. E. P.?

SOLUTION.— (*a*) Reducing the fractional parts of the sum of the lengths of the ordinates to a decimal, we have $\frac{5}{16}$ inch = .3125 inch. The length of the mean ordinate is, then, $12.3125 \div 10 = 1.23125$ inches, or to the nearest .001, 1.231 inches. Ans.

(*b*) Multiplying the length of the mean ordinate by the scale of the spring, the M. E. P. is $1.231 \times 20 = 24.62$ lb. per sq. in. Ans.

44. Locating the Ordinates.—The length of the diagram will seldom be divisible into equal parts that can readily be laid off by a scale, and to divide the length into equal parts by a cut-and-try process will be found very tedious. These difficulties may, however, be overcome by an application of a simple geometrical principle, in the manner illustrated in Fig. 12. The tangent lines *a b* and *c d* are first drawn perpendicular to the atmospheric line *m n*. A scale is then selected so graduated that when the *0* mark is placed on the line *a b* and the scale lies diagonally across the diagram, the desired number of spaces will be included between the *0* mark and a mark that will fall on the line *c d*. In Fig. 12 it was desired to divide the diagram so as to get 10 ordinates. The length of the diagram is a little less than 5 inches; a scale graduated in inches can, therefore, readily be placed with the *0* mark on the line *a b* and the 5-inch mark on the line *c d*. Lines drawn parallel to *a b* and *c d* through each of the inch and half-inch marks from *0* to *5* would evidently divide the diagram into 10 spaces of equal width, and since the ordinates are to be drawn through the middle of these spaces, we see that to locate the ordinates

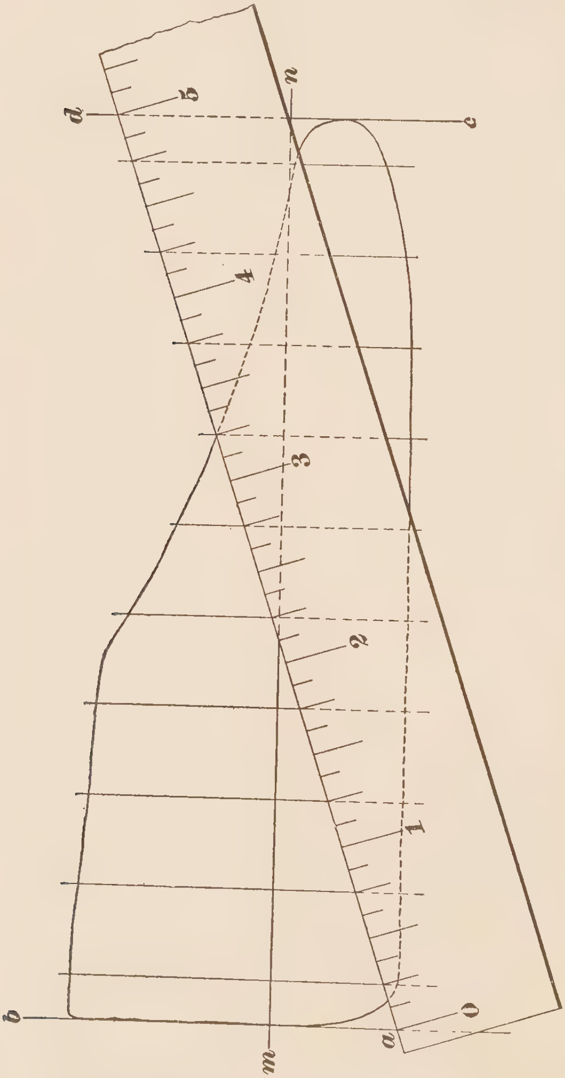

FIG. 12.

it is only necessary to make a mark on the diagram opposite each of the quarter-inch marks on the scale, and draw parallels to *a b* and *c d* through these marks.

45. Mean Ordinate of a Diagram With Loops.—To find the mean ordinate of a diagram with loops (see Fig. 11), subtract the sum of the lengths of the ordinates of the loops from the sum of the lengths of the ordinates of the main part of the diagram and divide by the total number of ordinates. In order to get reasonably accurate results with a diagram of this kind, it will generally be necessary to use a greater number of ordinates than are required for a more simple form of diagram.

46. Approximate Determination of M. E. P.—To approximately determine the M. E. P. of an engine when the point of apparent cut-off is known, and the boiler pressure, or the pressure per square inch in the boiler, from which the supply of steam is obtained, is given, and when an indicator diagram is not obtainable, use the following rule:

Rule 3.—*Add 14.7 to the gauge pressure and multiply the number opposite the fraction indicating the point of cut-off in the table, Art.* **46,** *by the pressure. Subtract 17 from the product and multiply by .9. The result is the M. E. P. for good, simple non-condensing engines.*

Cut-off.	Constant.	Cut-off.	Constant.	Cut-off.	Constant.
$\frac{1}{6}$	.545	$\frac{3}{8}$	.773	$\frac{2}{3}$	.943
$\frac{1}{5}$	.590	.4	.794	.7	.954
$\frac{1}{4}$	.650	$\frac{1}{2}$	.864	$\frac{3}{4}$	.970
.3	.705	.6	.916	.8	.981
$\frac{1}{3}$	.737	$\frac{5}{8}$	.927	$\frac{7}{8}$	.993

If the engine is a simple condensing engine, subtract the pressure in the condenser instead of 17. The fraction indicating the point of cut-off is obtained by dividing the distance that the piston has traveled when the steam is cut off by the whole length of the stroke; i. e., it is the apparent cut-off. For a $\frac{2}{3}$ cut-off and 92 pounds gauge pressure in the boiler, the M. E. P. is $[92 + 14.7) \times .943 - 17] \times .9 = 75.3$ pounds per square inch.

It is to be observed that this rule cannot be applied to a compound engine or any other engine in which the steam is expanded in successive stages in several cylinders.

Example.—Find the approximate M. E. P. of a non-condensing engine cutting off at $\frac{1}{2}$ stroke and making 240 revolutions per minute. The boiler pressure is 80 pounds gauge.

Solution.— $80 + 14.7 = 94.7$. Using rule **3** and table, Art. **46**, the constant for $\frac{1}{2}$ cut-off is .864, and $.864 \times$ boiler pressure $= .864 \times 94.7 = 81.82$. M. E. P. $= (81.82 - 17) \times .9 = 58.34$ lb. per sq. in. Ans.

EXAMPLES ON FINDING THE M. E. P.

Example 1.—The projection of the head-end diagram shown in Fig. 13 upon the atmospheric line is the distance *A Z*, and it is divided,

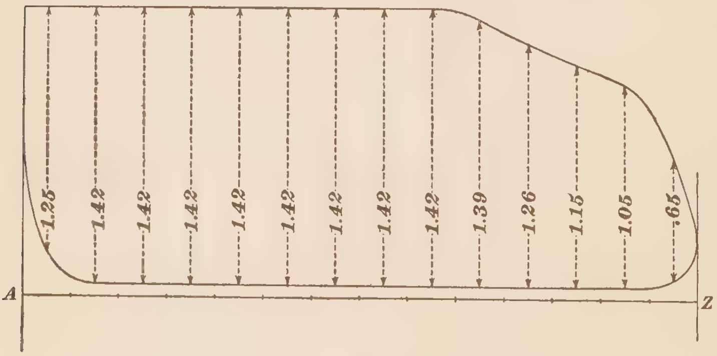

Fig. 13.

in this case into 14 equal spaces. The length of each of the perpendicular lines drawn through the diagram opposite the centers of these spaces is marked on the line itself. The scale of the spring used in obtaining the diagram was 40 pounds. Find (*a*) the length of the mean ordinate and (*b*) the M. E. P. of the diagram.

SOLUTION.— (*a*) The sum of the lengths of the 14 ordinates is 18.11 inches; the length of the mean ordinate is, therefore, 18.11 ÷ 14 = 1.294 in., nearly. Ans.

(*b*) Multiplying the length of the mean ordinate by the scale of the spring, we have M. E. P. = 1.294 × 40 = 51.76 lb. per sq. in. Ans.

EXAMPLE 2.—The projection of the crank-end diagram shown in Fig. 14 upon the atmospheric line is the distance *A Z*, and it is divided in this case into 14 equal spaces. The length of each of the perpendicular lines drawn through the diagram opposite the centers of these spaces is marked on the line itself. The scale of the spring is 40 pounds. Find (*a*) the mean ordinate and (*b*) the M. E. P. of the diagram.

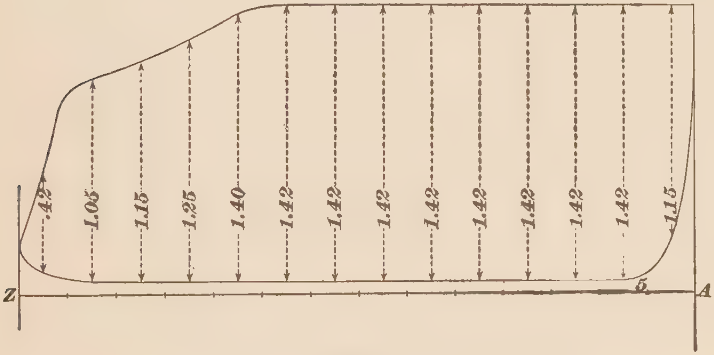

FIG. 14.

SOLUTION.— (*a*) The sum of the lengths of the 14 ordinates is 17.78 inches; the length of the mean ordinate is, therefore, 17.78 ÷ 14 = 1.27 in. Ans.

(*b*) Multiplying the mean ordinate by the scale of the spring, we have M. E. P. = 1.27 × 40 = 50.8 lb. per sq. in. Ans.

EXAMPLE 3.—What was the average M. E. P. in the cylinder during the revolution represented by the two diagrams in examples 1 and 2?

SOLUTION.—Since the M. E. P. in the head end was 51.76 pounds per square inch and that in the crank end was 50.8 pounds per square inch, the average for the two strokes making up the complete revolution was $\frac{51.76 + 50.8}{2} = 51.28$ lb. per sq. in. Ans.

CALCULATING THE INDICATED HORSEPOWER.

47. General Rule for Calculating I. H. P.—Knowing the dimensions and speed of the engine and the mean effective pressure on the piston, we have all the data for finding the rate of work done in the engine cylinder expressed in horsepowers. Work is the product of force multiplied by the distance through which it acts. In the case of the engine cylinder, the total force is the M. E. P. per square inch multiplied by the area of the piston; and the distance through which the force acts in 1 minute is the distance the piston moves in 1 minute, which is equal to the number of strokes per minute multiplied by the length of the stroke.

Rule 4.—*To find the indicated horsepower developed by the engine, multiply together the M. E. P. per sq. in., the area of piston in square inches, the length of stroke in feet, and the number of strokes per minute. Divide the product by 33,000; the result will be the indicated horsepower of the engine.*

Let I. H. P. = indicated horsepower of engine;
P = M. E. P. in pounds per square inch;
A = area of piston in square inches;
L = length of stroke in feet;
N = number of working strokes per minute.

Then, the above rule may be expressed thus:

$$\text{I. H. P.} = \frac{PLAN}{33,000}.$$

In a double-acting engine the number of working strokes per minute is twice the number of revolutions per minute. For example, if a double-acting engine runs at a speed of 210 revolutions per minute there are 420 working strokes per minute. A few types of engines, however, are single-acting; that is, the steam acts on only one side of the piston. Such are the Westinghouse, the Willans, and others. In this case, only one stroke per revolution does work, and, consequently, the number of strokes per minute to be used in the above rule is the same as the number of revolutions per minute.

Unless it is specifically stated that an engine is single-acting, it is always understood, when the dimensions of an engine are given, that a double-acting engine is meant.

48. Piston Speed.—The product LN of rule 4 gives the total distance traveled by the piston in 1 minute. This is called the **piston speed.** It is usual to take the stroke in inches. Then, to find the piston speed, multiply the stroke in inches by the number of strokes and divide by 12, or, letting S represent the piston speed, $S = \frac{lN}{12}$, where l is the stroke in inches. But $N = 2R$, where R represents the number of revolutions per minute. Hence,

$$S = \frac{lN}{12} = \frac{l \times 2R}{12} = \frac{lR}{6}.$$

Rule 5.—*To find the piston speed of an engine, multiply the stroke in inches by the number of revolutions per minute and divide the product by 6.*

Example.—An engine with a 52-inch stroke runs at a speed of 66 revolutions per minute. What is the piston speed?

Solution.—By rule 5, $S = \frac{52 \times 66}{6} = 572$ ft. per min. Ans.

The piston speeds used in modern practice are about as follows:

	Ft. per min.
Small stationary engines........	300 to 600.
Large stationary engines........	600 to 1,000.
Corliss engines..................	400 to 750.
Marine engines..................	200 to 1,200.

49. Allowance for Area of Piston Rod.—It is generally considered sufficiently accurate to take the total area of one side of the piston as the area to be used in calculating the horsepower of an engine. The effective area of one side of the piston is, however, reduced by the sectional area of the piston rod, and if it is important that the power be calculated to the greatest practicable degree of accuracy, an allowance for the area of the piston rod must be made. This is done by taking as the piston area one-half the sum of the areas exposed to steam pressure on the two sides of the piston. Thus, if we have a piston 30 inches diameter with a 6-inch piston rod, the average area is $\frac{30^2 \times .7854 + (30^2 \times .7854 - 6^2 \times .7854)}{2}$ $= 692.72$ square inches. If the piston rod is continued past the piston so as to pass through the head-end cylinder head, i. e., if the piston has a **tail rod,** allowance must be made for the tail rod. Thus, with a piston 30 inches diameter, a piston rod 6 inches diameter, and a tail rod 5 inches diameter, the average area is $\frac{(30^2 \times .7854 - 5^2 \times .7854) + (30^2 \times .7854 - 6^2 \times .7854)}{2}$ $= 682.9$ square inches.

EXAMPLE 1.—The diameter of the piston of an engine is 10 inches and the length of stroke 15 inches. It makes 250 revolutions per minute with an M. E. P. of 40 pounds per square inch. What is the horsepower?

SOLUTION.—The number of working strokes is $250 \times 2 = 500$ per minute. Applying rule **4**, we get

$$\text{I. H. P.} = \frac{40 \times \frac{15}{12} \times (10^2 \times .7854) \times 500}{33{,}000} = 59.5 \text{ H. P.} \quad \text{Ans.}$$

EXAMPLE 2.—In Fig. 15 are shown two indicator diagrams taken from an 18″ × 20″ engine, making 200 revolutions per minute. The scale of the spring is 60. Compute (*a*) the mean effective pressure and (*b*) the horsepower.

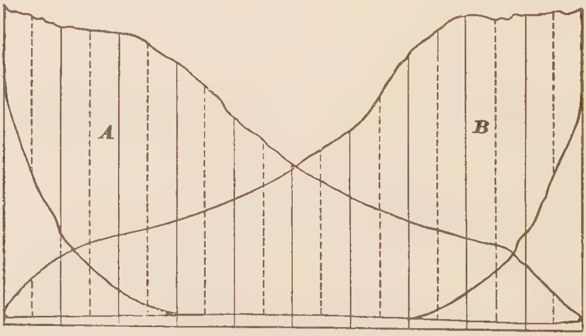

FIG. 15.

SOLUTION.—(*a*) Divide the diagrams into 10 equal parts, as shown by the full lines. Then, as previously directed, draw lines or ordinates (see dotted lines in cut) perpendicular to the atmospheric line through the middle points of each of the 10 equal divisions. Measuring the lengths of all the dotted lines and adding them together, we find the sum of the lengths for diagram A is 7.8 inches and for diagram B 7.84 inches. Dividing each of these results by 10 and multiplying by the scale of the spring, we have $\frac{7.8}{10} \times 60 = 46.8$ pounds per square inch = M. E. P. for diagram A, and $\frac{7.84}{10} \times 60 = 47.04$ pounds per square inch = M. E. P. for diagram B. The average M. E. P. for both cards is $\frac{46.8 + 47.04}{2} = 46.92$ lb. per sq. in. Ans.

(*b*) To find the horsepower, the value for the M. E. P. must be substituted for P in the formula corresponding to rule 4, $\frac{PLAN}{33{,}000} = \text{H. P.}$ Reducing the stroke to feet, and substituting the values of P, L, A, and N, we have

$$\frac{46.92 \times \frac{20}{12} \times (18^2 \times .7854) \times (200 \times 2)}{33{,}000} = 241.21 \text{ H. P.} \quad \text{Ans.}$$

ENGINE CONSTANTS.

50. An **engine constant** for a given engine is a number obtained by combining into a single factor all the factors of the horsepower rule that are constant for that engine. This factor may then be substituted for the factors that were combined to produce it, and a new rule obtained for that engine, in which the number of unknown quantities is less than in the original rule. The labor involved in calculating the I. H. P. for the engine is thus considerably reduced.

51. Constant for a Uniform Speed of Rotation.—When the speed of rotation of a given engine is uniform, all the factors except the mean effective pressure are constant; the engine constant for this case can, therefore, be found by the following rule:

Rule 6.—*Multiply together the length of the stroke in feet, the area of the piston in square inches, and the number of* *working strokes per minute, and divide the product by 33,000; the quotient will be the engine constant.*

This rule may be expressed by the formula

$$C_u = \frac{L\,A\,N}{33,000};$$

in which C_u is the engine constant for the uniform speed of rotation, and L, A, and N have the same meaning as in the formula corresponding to rule **4.** The constant C_u is the horsepower of the engine for a mean effective pressure of 1 pound per square inch.

To find the I. H. P. when the engine constant for a uniform speed of rotation is known, multiply the engine constant by the M. E. P.

Example 1.—What is the engine constant for a 16″ × 20″ engine running at a uniform speed of 200 R. P. M.?

Solution.—The length L of the stroke is $\frac{20}{12}$ feet, the area A of the piston is $16^2 \times .7854 = 201$ square inches, and the number of strokes N is $2 \times 200 = 400$. Substituting these values in the formula corresponding to rule **6,** we have

$$C_u = \frac{\frac{20}{12} \times 201 \times 400}{33,000} = 4.06. \quad \text{Ans.}$$

Example 2.—What is the I. H. P. of the engine of example 1 when the average M. E. P. for a pair of indicator diagrams is 43.2 pounds per square inch?

Solution.—Multiplying the engine constant by the M. E. P., we have I. H. P. $= 4.06 \times 43.2 = 175.39$. Ans.

52. Constant for a Varying Speed of Rotation.—When the speed of rotation is variable, the engine constant is given by the following rule:

Rule 7.—*Multiply together twice the length of stroke in feet and the area of the piston in square inches; divide the product by 33,000 for a double-acting engine. For a single-acting engine, multiply the length of stroke in feet by the area of the piston in square inches and divide the product by 33,000.*

Or, $C_v = \dfrac{2\,L\,A}{33,000}$ for double-acting engines, and $C_v = \frac{L A}{33,000}$ for single-acting engines,

where C_v = engine constant.

The value of C_v derived from these formulas is the I. H. P. of the engine for a speed of 1 revolution per minute and a mean effective pressure of 1 pound per square inch. To find the I. H. P., multiply this constant by the number of revolutions per minute and the product so obtained by the M. E. P.

53. Formulas for M. E. P. and I. H. P. in Terms of Area of Diagram.—The fact that the M. E. P. of a diagram is equal to its area in square inches divided by its length in inches and this product multiplied by the scale of the spring enables us to develop a formula by means of which the horsepower can be calculated from the area and length of the diagram and a constant that is obtained by multiplying the engine constant by the scale of the spring. Such a formula will be found convenient when the area of the diagram is measured by a planimeter that cannot be set to measure the M. E. P. of the diagram directly.

Let a = area of diagram in square inches;
l = length of diagram in inches;
s = scale of spring.

Then $$\text{M. E. P.} = \frac{a s}{l}.$$

This value of M. E. P. can be substituted for P in the formula corresponding to rule **4**, giving us the formula

$$\text{I. H. P.} = \frac{\frac{a s}{l} L A N}{33,000}.$$

For a given engine from which a number of diagrams are to be taken, the factors s, L, A, and N will generally be constant; these factors may, therefore, be combined with the factor 33,000 in the same manner as was done in Art. **51**; a constant which we will call C_a may thus be obtained which will be given by

Rule 8.—*Multiply together the scale of the indicator spring, the length of stroke in feet, the area of the piston in square inches, and the number of working strokes per minute. Divide the product by 33,000.*

Or, $$C_a = \frac{s\,L\,A\,N}{33{,}000}.$$

The indicated horsepower will then be given by multiplying this constant by the area of the diagram if the engine is single-acting, or the average area of the two diagrams if the engine is double-acting, and dividing the product by the length of the diagram.

54. If the indicator reducing motion is so constructed that the length l of the diagrams is constant, the constant may be made to include this factor. Calling such a constant C_l, we have

Rule 9.—*Multiply together the scale of the indicator spring, the length of stroke in feet, the area of the piston in square inches, and the number of working strokes per minute. Divide this product by the product of 33,000 and the length of the diagram.*

Or, $$C_l = \frac{s\,L\,A\,N}{33{,}000\,l}.$$

With this constant, the indicated horsepower can be found by multiplying it by the area of the diagram if the engine is single-acting, or the average area of the two diagrams when the engine is double-acting.

EXAMPLE.—Calculate the value of the constant by which to multiply the area of the diagrams to find the I. H. P. for a 12″ × 16″ engine running at 250 R. P. M. when the scale of the spring is 50 and the length of the diagrams is 3½ inches.

SOLUTION.—The length L of the stroke is $\frac{16}{12}$ feet, the area A of the piston is $12^2 \times .7854 = 113.1$ square inches, and the number N of working strokes is $2 \times 250 = 500$ per minute. Substituting these values and the given values for the scale of the spring and the length of the diagram in rule **9**, we have

$$C_l = \frac{50 \times \frac{16}{12} \times 113.1 \times 500}{33{,}000 \times 3\frac{1}{2}} = 32.64. \quad \text{Ans.}$$

55. Formula for I. H. P. in Terms of Total Length of Ordinates.—

Let n = number of ordinates drawn on diagram;
o = sum of the lengths of ordinates in inches;
h = length of mean ordinate in inches;
C_o = constant for calculating the I. H. P. from the ordinates;
s = scale of indicator spring.

In accordance with Art. **43,** the length of the mean ordinate is equal to the sum of the lengths of the ordinates divided by their number; that is,

$$h = \frac{o}{n};$$

and in accordance with Art. **42,** the mean effective pressure is equal to the length of the mean ordinate multiplied by the scale of the spring, or

$$\text{M. E. P.} = s\,h = s\frac{o}{n}.$$

Substituting this value of the M. E. P. for P in the formula corresponding to rule **4,** we have

$$\text{I. H. P.} = \frac{s\frac{o}{n}L\,A\,N}{33{,}000}.$$

56. For the diagrams taken from a given engine running at a uniform rate of speed, the factors s, n, L, A, and N are generally constant. They may, therefore, be combined with the constant factor 33,000 to form a new constant whose value is given by the following rule:

Rule 10.—*Multiply together the scale of the indicator spring, the length of stroke in feet, the area of the piston in square inches, and the number of working strokes per minute. Divide this product by the product of 33,000 and the number of ordinates.*

Or,

$$C_o = \frac{s\,L\,A\,N}{33{,}000\,n}.$$

This constant multiplied by the sum o of the lengths of the ordinates in inches for a diagram gives the indicated horsepower for a single-acting engine. For a double-acting engine one-half the sum of the lengths of the ordinates of the two diagrams is to be taken. It is to be observed that the number of ordinates must be the same for each diagram, and that in case of a double-acting engine the *sum* of the number of ordinates of the two diagrams *must not be used.*

EXAMPLE 1.—Calculate the value of the constant C_o for diagrams taken with a 40 spring from a 28″ × 42″ engine running at 90 R. P. M. when the number of ordinates is 20.

SOLUTION.—The area A of the piston is $28^2 \times .7854 = 615.75$ square inches; the length L of the stroke is $\frac{42}{12} = 3\frac{1}{2}$ feet, and the number N of working strokes is $2 \times 90 = 180$ per minute. Substituting these and the values given for the scale s of the spring and the number n of ordinates in rule **10,** we have

$$C_o = \frac{40 \times 3\frac{1}{2} \times 615.75 \times 180}{33{,}000 \times 20} = 23.51. \quad \text{Ans.}$$

EXAMPLE 2.—What is the I. H. P. of the engine of example 1, when one-half the sum of the lengths of the 20 ordinates of the two diagrams is $19\frac{3}{16}$ inches?

SOLUTION.—

$$\text{I. H. P.} = 23.51 \times 19\tfrac{3}{16} = 451.1, \text{ nearly.} \quad \text{Ans.}$$

BRAKE HORSEPOWER.

DYNAMOMETERS.

57. Dynamometers are instruments for measuring power. They are divided into two main classes: *absorption dynamometers* and *transmission dynamometers.*

58. The most common form of **absorption dynamometer** is the Prony brake, which consists simply of a friction brake designed to absorb in friction and measure the work done by a motor, or the power given out by a shaft.

59. A **transmission dynamometer** is used to measure the power required to drive a machine or do other work; then, to determine the power required to run the shafting in a mill, a transmission dynamometer would be interposed between the shafting and the source of power, and by suitable belt connections the shafting would be driven through the dynamomoter, from which the power could be determined. Since transmission dynamometers do not enter into the work of the steam engineer, they will not be treated of here.

60. **Brake horsepower** is a term often applied to the power measured by a Prony brake or other type of absorption dynamometer. The brake horsepower of an engine or other motor working under given conditions is the same as the net horsepower. Since the power measured by an absorption dynamometer is the power a motor delivers at the shaft or flywheel, it is sometimes called the **delivered horsepower.**

61. **The Prony Brake.**—Fig. 16 represents a simple and common form of Prony brake. It consists of two wooden blocks *A* and *B* that are clamped together upon a pulley *P* by the bolts and thumbnuts *c*, *c*. The same bolts clamp an arm *L* to the upper block, from which a scale pan bearing a known weight *W* is suspended. The distance *R* from the center of the pulley to the perpendicular through the point from which the scale pan is suspended is also known. The counterweight *w* should be so adjusted as to just balance the extra length of *L* on the right and the weight of the scale pan.

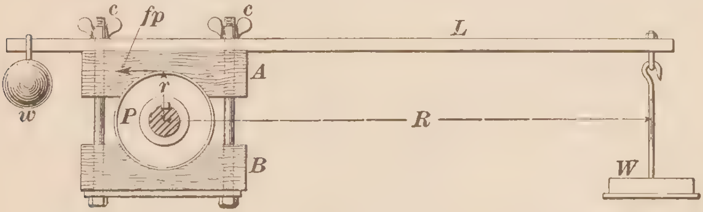

FIG. 16.

Suppose the pulley to revolve left-handed and the bolts c, c tightened until, with a weight W in the scale pan, the lever L will remain stationary in a horizontal position. Then the foot-pounds of work absorbed by the brake can be found by multiplying the weight W by the circumference of a circle whose radius is R (in feet) and by the number of revolutions of the pulley.

Rule 11.—*To find the horsepower, multiply the weight in the scale pan by the length in feet of the lever arm about the center of the shaft, by the number of revolutions of the pulley per minute, and by 6.2832. Divide the product by 33,000.*

Or,
$$\text{H. P.} = \frac{W R N \times 6.2832}{33,000},$$

where H. P. = number of horsepower absorbed;
R = length in feet of lever arm about center of shaft;
W = weight in scale pan;
N = number of revolutions per minute.

Example.—A brake with an arm R 6 feet long was placed on the flywheel of an engine. If the engine ran at 200 revolutions per minute, what power did it develop when the brake balanced with 14 pounds in the scale pan?

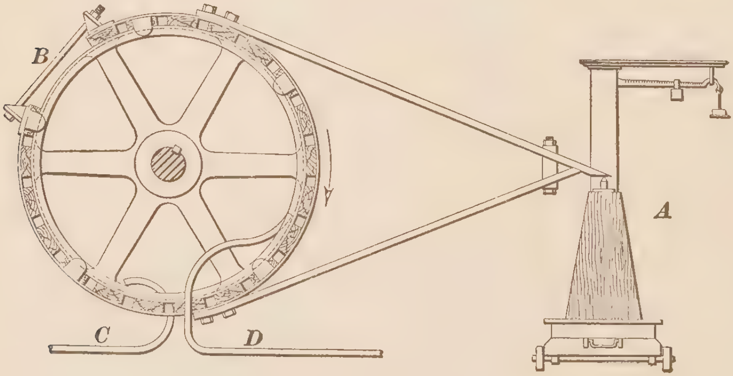

Fig. 17.

Solution.—Applying the rule just given, we get

$$\text{H. P.} = \frac{14 \times 6 \times 200 \times 6.2832}{33,000} = 3.198. \quad \text{Ans.}$$

62. **Brakes** are often constructed of a metal band that extends entirely around the pulley, the rubbing surface being formed of blocks of wood fitted to the inside of the band. A weight arm is attached to one side of the pulley, and the friction is varied by means of a bolt and nut used to connect the two ends of the band.

Instead of hanging weights in the scale pan, the friction may be weighed on a platform scale, as shown in Fig. 17. In this case, the direction of rotation of both pulley and arm is the same. Rule **11** may be used for calculating the brake horsepower, substituting the weight indicated by the scale for the weight in the scale pan, and taking the length of the lever as the distance between the center of the shaft and the point where the lever presses on the platform. In

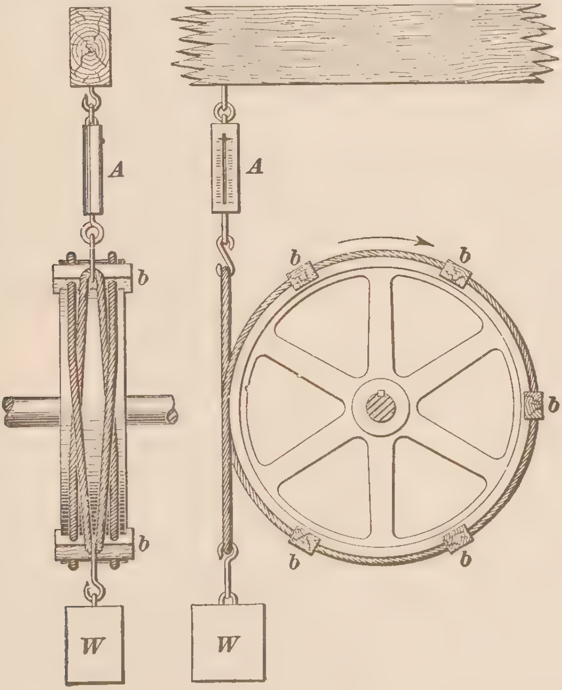

FIG. 18.

reading the weight off the scale beam, it must be remembered that the weight to be used in the calculation is the *difference* between the weight at which the scale balances when the brake is not applied and when applied.

63. It is essential that Prony brakes should be well lubricated, and for all except small powers, means must be provided for conducting away the heat generated by friction. If there are internal flanges on the brake wheel, water can be run on the inside of the rim, the flanges serving to retain the water at the sides and centrifugal force to keep it in contact with the rim. A funnel-shaped scoop can be used to remove the water. It should be attached to a pipe and placed so as to scoop out the water, which should flow continuously. This arrangement is shown in Fig. 17.

64. A **rope brake,** like that in Fig. 18, will give good results. The figure shows the construction so clearly that no description is necessary. To obtain the brake load, subtract the brake pull as registered by the spring balance from the weight. In this case, the lever arm is equal to the radius of the pulley plus one-half the diameter of the rope, expressed in feet.

STEAM CONSUMPTION OF SIMPLE ENGINES.

CALCULATIONS RELATING TO STEAM CONSUMPTION.

65. The indicator diagram also enables us to find approximately the amount of steam consumed by the engine. In referring to the steam consumption, it is customary to take as a unit the *steam consumed per horsepower per hour.* It is to be observed that the expressions "steam consumption" and "water consumption" when applied to a steam engine are synonymous.

Take a point *a* on the expansion line before the release (see Fig. 19); measure the pressure from the vacuum line, and from column 6 of the Steam Table find the weight of a cubic foot at that pressure. The cubic contents of the cylinder (including the clearance) up to the point *a* multiplied by the weight per cubic foot, must give the weight of steam in the cylinder at this instant. This weight would be the steam consumed per stroke were it not for two circumstances. (1) When the fresh steam from the boiler enters the cylinder, it comes in contact with the cylinder walls, which have been cooled down by the exhaust steam. A glance at the Steam Table shows that the exhaust steam is at a great deal lower temperature than the fresh steam. Consequently, part of the incoming steam condenses, and, of course, the indicator diagram takes no account of this condensed steam. Hence, the steam actually in the cylinder is less than originally entered from the boiler, because part of the original steam has changed to water. (2) On account of the closure of the exhaust port, some steam is compressed and saved.

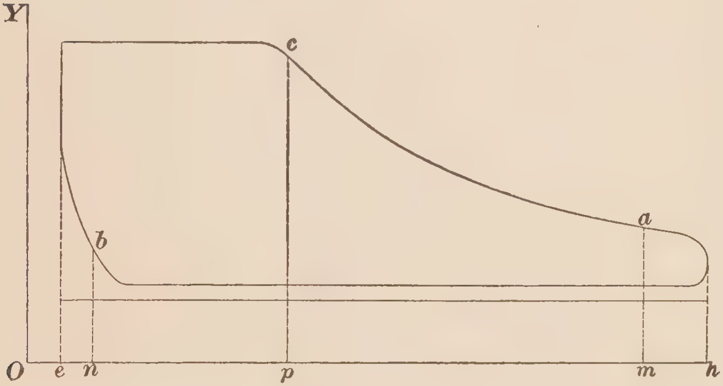

FIG. 19.

66. To find the weight of the steam saved by compression, take a point *b* on the compression curve, measure its pressure from vacuum as before, and compute the weight of the steam in the cylinder up to *b*. Subtract this from the weight first obtained, and the difference will be the weight of steam per stroke accounted for by the indicator. Multiply this weight per stroke by the number of strokes per hour and divide by the I. H. P. of the engine. The result will be the steam used per I. H. P. per hour.

Example.—Fig. 19 represents an indicator diagram taken with a 45 spring from an engine having an 18″ × 24″ cylinder, running at 120 revolutions and developing 130 horsepower. The clearance is 5 per cent. Find the steam consumption per I. H. P. per hour.

Solution.—Project the two ends of the diagram perpendicularly upon the vacuum line $O\,h$, as at e and h; $e\,h$ is then the length of the diagram. Lay off $e\,O$ equal to the clearance; that is, equal to 5 per cent. of $e\,h$. Draw $O\,Y$ perpendicular to $O\,h$. Take the point a near the point of release and measure the distances $a\,m$ and $O\,m$. Take the point b somewhere on the compression line and measure the distances $b\,n$ and $O\,n$. The measurements are found to be:

$$\begin{aligned} a\,m &= .71 \text{ inch;} \\ O\,m &= 3.17 \text{ inches;} \\ b\,n &= .6 \text{ inch;} \\ O\,n &= .333 \text{ inch.} \end{aligned}$$

The length of the diagram $= e\,h = 3\frac{1}{3}$ inches; the length of the stroke is 2 feet. Hence, each inch of the length of the diagram equals $\frac{2}{3\frac{1}{3}} = .6$ foot of stroke. Since the scale of the indicator spring is 45, the above measurements reduced to pressures in pounds per square inch and feet of stroke become:

$$\begin{aligned} a\,m &= .71 \times 45 = 31.95 \text{ pounds;} \\ b\,n &= .6 \times 45 = 27 \text{ pounds;} \\ O\,m &= 3.17 \times .6 = 1.9 \text{ feet;} \\ O\,n &= .333 \times .6 = .2 \text{ foot.} \end{aligned}$$

The area of the piston is $18^2 \times .7854 = 254.47$ square inches $= \frac{254.47}{144} = 1.767$ square feet. Consequently, the volume of steam in the cylinder when the piston is at the point represented by a is $1.9 \times 1.767 = 3.3573$ cubic feet. The volume when the piston is at b is $.2 \times 1.767 = .3534$ cubic foot. The weight of a cubic foot of steam at an absolute pressure of 31.95 pounds per square inch is found from the Steam Table to be .078723 pound; and at a pressure of 27 pounds, the weight is .067207 pound. Hence, the weight of the steam in the cylinder is $.078723 \times 3.3573 = .264297$ pound; while the weight of steam saved by compression is $.067207 \times .3534 = .023751$ pound. The steam used per stroke is, therefore, $.264297 - .023751 = .240546$ pound. To find the amount used per I. H. P. per hour, multiply the weight used per stroke by the number of strokes per hour and divide by the I. H. P. Therefore, the required weight is

$$\frac{.240546 \times 120 \times 2 \times 60}{130} = 26.645 \text{ lb. Ans.}$$

67. Suppose the weight of the steam in the cylinder to be calculated by taking the point c near the point of cut-off. $c p = 1.59$ inches $= 1.59 \times 45 = 71.55$ pounds; $O p = 1\frac{1}{3}$ inches $= \frac{4}{3} \times .6 = .8$ foot of stroke. The volume of steam in the cylinder when the piston is at c is, therefore, $.8 \times 1.767 = 1.4136$ cubic feet. One cubic foot of steam at the pressure of 71.55 pounds, absolute, weighs .168009 pound. The weight of the steam in the cylinder is, therefore, $.168009 \times 1.4136 = .237498$ pound. Subtracting the steam saved by compression, the steam used per stroke is $.237498 - .023751 = .213747$ pound, and the steam per I. H. P. per hour is

$$\frac{.213747 \times 120 \times 2 \times 60}{130} = 23.677 \text{ pounds.}$$

Now, unless the valve leaks, the weight of the steam when the piston is at a can be no greater than when it is at c, since no fresh steam has been allowed to enter; but the calculation shows that there is .264297 pound in the cylinder when the piston is at a, and only .237498 pound when the piston is at c. This shows that $.264297 - .237498 = .026799$ pound has been condensed to water by the time the piston has arrived at c, but has been re-evaporated before the piston arrives at a. Hence, by calculating the water consumption at cut-off and then at release, a good idea of the amount of cylinder condensation may be obtained. If the steam used by the engine be actually caught and weighed and then compared with the weight as calculated from release, an idea may be obtained of the amount of condensation at release. The computed consumption is always less than the actual consumption.

68. Where there is a sufficient amount of compression, the work may be simplified by taking the two points a and b at the same height above the vacuum line, as shown in

Fig. 20. Since the absolute pressure at a and b is the same, the clearance may be left entirely out of account. and the volume to be used in the computation will be $\frac{l}{L}$ times the volume of the cylinder, or, in other words, $\frac{l}{L} \times$ length of stroke × area of piston. When this method is used, the steam consumption may be found directly from the formula

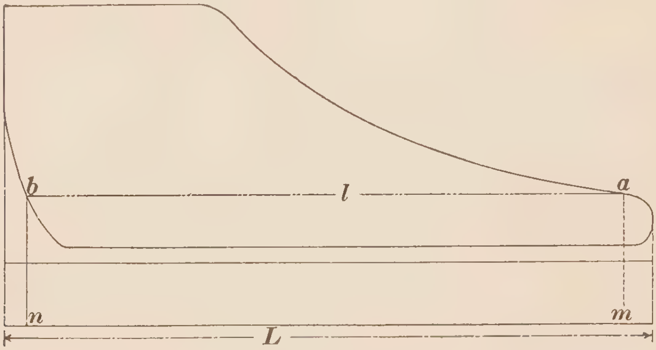

FIG. 20.

$$Q = \frac{13{,}750\, l\, W}{P L},$$

in which Q is the number of pounds of steam consumed per horsepower per hour; W, the weight of a cubic foot of steam at the absolute pressure a, and P, the M. E. P.

Expressing the formula in words, we have the following rule:

Rule 12.—*Take two points, one on the expansion line and one on the compression line, both equally distant from the vacuum line. Find the pressure of the steam at these points, and from the Steam Table find the weight of a cubic foot of steam at that pressure. Multiply this weight by the distance between the two points and by 13,750. Divide the product by the M. E. P. and by the length of the diagram. The result will be the pounds of steam consumed per I. H. P. per hour, as shown by the diagram.*

EXAMPLE.—From a diagram taken from an $18\frac{1}{2}'' \times 30''$ engine, the following measurements were obtained (see Fig. 20): $am = .667$ inch; $l = 3.08$ inches; $L = 3.5$ inches; M. E. P. = 35 pounds; spring, 45. What is the steam consumption per I. H. P. per hour?

SOLUTION.—The indicator diagram being taken with a 45 spring, the pressure at a is $45 \times .667 = 30$ pounds, absolute. The weight of a cubic foot of steam at this pressure is .0742 pound. Using rule **12,**

$$Q = \frac{13{,}750 \times 3.08 \times .0742}{35 \times 3.5} = 25.65 \text{ lb.} \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. Size of engine; $12'' \times 20''$; length of diagram L, 3.4 inches; length l, $2\frac{1}{4}$ inches; height am, $\frac{3}{8}$ inch; R. P. M., 230; spring, 30; M. E. P., 18 pounds per square inch. What is the steam consumption per I. H. P. per hour? Ans. 25.63 lb. per I. H. P. per hr.

2. Size of engine, $12'' \times 12''$; M. E. P., 51.1; length of diagram L, 2.6 inches; length l, 1.8 inches; height am, .7 inch; R. P. M., 350; spring, 70. What is the steam consumption per I. H. P. per hour? Ans. 21.92 lb. per I. H. P. per hr.

3. If, in the above engine, example 2, the pressure at cut-off is 110 pounds, absolute; the clearance is 8 per cent.; the length of the diagram to the point of cut-off is .7 inch; the pressure at a point on the compression curve is 49 pounds, absolute, and the distance of this point from the end of the diagram is .14 inch, what is the steam consumption per I. H. P. per hour at cut-off? Ans. 19.44 lb. per I. H. P. per hr.

SIZE OF STEAM ENGINES.

69. The problem of selecting a size of simple engine that will develop a given indicated horsepower is capable of an infinite number of correct solutions, depending on the conditions present. The factors that determine the indicated horsepower are the mean effective pressure, the length of stroke, the diameter of the piston, and the number of revolutions per minute. Before the diameter of the piston and the length of stroke, which data constitute the size of the engine, can be determined, the boiler pressure, point of cut-off, and piston speed must be chosen. From the boiler pressure and the point of cut-off the mean effective pressure is then estimated in the manner explained in Art. **46.** The area of the piston is then given by the following rule:

Rule 13.—*To find the piston area in square inches, multiply the indicated horsepower by 33,000 and divide by the product of the mean effective pressure and the piston speed in feet per minute.*

Or,
$$A = \frac{33,000\,H}{PS},$$

where
$A =$ area of piston;
$H =$ indicated horsepower;
$P =$ mean effective pressure;
$S =$ piston speed.

To find the diameter, divide the result of rule **13** by .7854 and extract the square root of the quotient.

EXAMPLE.—Find the piston diameter for a 25-horsepower engine using steam at 70 pounds gauge pressure, cutting off at ¾ stroke, and to have a piston speed of 300 feet per minute. Engine is non-condensing.

SOLUTION.—By rule **3**, Art. **46**, the mean effective pressure is $[(70 + 14.7) \times .937 - 17] \times .9 = 56$ pounds per square inch. Applying rule **13**, we get

$$A = \frac{33,000 \times 25}{56 \times 300} = 49.1 \text{ square inches.}$$

The corresponding diameter is $\sqrt{\frac{49.1}{.7854}} = 8$ in., about. Ans.

70. Since the piston speed is the product of the number of strokes and the length of stroke, to find the latter the number of strokes must be assumed. Then, to find the length of stroke in feet, divide the piston speed by the number of strokes.

EXAMPLE.—If the engine in the example given in Art. **69** is to make 120 revolutions per minute, what should be the stroke in inches?

SOLUTION.—Stroke in feet $= \frac{300}{120 \times 2} = 1.25$ feet, or $1.25 \times 12 = 15$ in. Ans.

GOVERNORS.

CONTROLLING THE SPEED OF AN ENGINE.

PURPOSE OF GOVERNORS.

1. The work to be done by an engine during a single stroke, or in other words the average resistance offered by the piston to the force acting on it, varies from time to time; the pressure of the steam in the boiler, and, in consequence, the force that may be made to act on the piston is also variable. In order to keep the speed of the engine practically uniform and to prevent its "racing," or running away," or its slowing down and stopping, it is evident that some means must be provided for so controlling the supply of steam admitted to the cylinder that the average force acting on the piston will be just great enough to overcome the average resistance. This duty is performed by the governor.

DUTY OF THE FLYWHEEL.

2. During each revolution of the engine there are variations in the force acting on the piston that cannot be directly controlled by the governor. At the beginning of the stroke the force is much greater than it is near the end, after the

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pressure has been reduced by the expansion of the steam. Furthermore, the turning effect on the crank of the force acting on the piston also varies. Since the resistance to be overcome at the circumference of the belt pulley or the resistance to be overcome by the engine shaft is nearly uniform for any single revolution of the shaft, it is evident that in order to keep the speed nearly uniform during a revolution, part of the energy when the turning effect is greatest must be stored so as to be given up again when the crank is near the dead centers and the pressure on the piston has fallen below the average pressure required to do the work. This duty is performed by the flywheel. When the turning effect on the crank is greater than the average turning resistance, a part of the energy is absorbed in increasing the speed of the flywheel rim; when the turning effect is less than the resistance, the flywheel rim gives up a part of the energy represented by its velocity and the speed falls. By making the diameter of the flywheel large and the rim heavy, the variation in speed required for the energy stored and given out during a single revolution is made so small as to be unobjectionable.

The flywheel also serves as a reservoir for storing the energy represented by the work done in the cylinder when there is a sudden drop in the load on the engine that cannot be immediately taken care of by the governor, or for furnishing a supply of energy to do the work when there is a sudden increase in the load. It thus aids in preventing too great a change in speed when there is a sudden change in the load.

METHODS OF CONTROLLING THE WORK DONE IN THE CYLINDER.

3. When the speed of the engine is uniform, the work done in the cylinder can be regulated only by varying the average net pressure on the piston. It is evident that this average pressure can be varied by three different methods: First, by varying the pressure of the steam admitted to the cylinder; second, by changing the point of cut-off so as to vary the ratio of expansion; third, by varying the back pressure or the pressure opposing the piston motion. Of these three methods, the first and second are the ones most generally used, although the third is often used in conjunction with the second.

GOVERNING BY THROTTLING.

4. In the first method, the flow of steam to the steam chest is controlled by a valve that regulates the pressure in the steam chest without affecting the expansion of the steam in the cylinder. The valve **throttles** the steam, that is, it obstructs its flow to a greater or less extent; this method of governing is, therefore, called **throttling,** and the device for automatically controlling the flow is called a **throttling governor.**

When steam has its pressure reduced by flowing through a narrow passage, like a partly closed valve. it is often said to be **wiredrawn.**

GOVERNING BY VARYING POINTS OF CUT-OFF AND COMPRESSION.

5. In the second and third methods of governing, the pressure of the steam admitted to the cylinder at the beginning of each stroke is nearly the same, and the work done in the cylinder is varied by controlling the action of the valve so as to cut off the steam supply earlier or later in the stroke. In many cases, the point at which compression begins is also varied, which has the effect of varying the back pressure or the resistance to be overcome by the piston. If the cut-off is regulated by hand, as, for example, is the case with the link motion of a locomotive, the engine is said to have an **adjustable cut-off.** If the action of the valve is automatically controlled, the device for regulating the cut-off is called an **automatic cut-off governor.** An engine with an automatic cut-off governor is called an **automatic cut-off engine.**

FORMS OF GOVERNORS.

THE CENTRIFUGAL GOVERNOR.

6. General Principles.—Most steam-engine governors depend for their action on the effect that the centrifugal force that is developed in a weight revolving around an axis outside of its center of gravity has in overcoming a varying resisting force. The weight is so suspended that the centrifugal force developed by its rotation is opposed by a variable resistance. When the speed and centrifugal force increase, the weight moves outwards and the resistance increases until it balances the increased centrifugal force; when the speed decreases, the centrifugal force becomes less than the resistance and the weight is forced towards the axis; as the weight moves towards the axis, the resistance decreases until it is again balanced by the centrifugal force in the new position. This motion of the weight is transmitted by a suitable mechanism to the valve that controls the flow of steam to the cylinder.

THE PENDULUM OR FLYBALL GOVERNOR.

7. The Revolving Pendulum.—The earliest and one of the simplest steam-engine governors is an application of what is scientifically known as the revolving pendulum (see Fig. 1). It consists of a weight *B* suspended from the point *O* by a fine cord and revolves around the vertical axis *O C*. When the pendulum revolves about the axis at a uniform speed, the ball remains at a constant distance *r* from the axis and at a constant distance *O C* below the point of suspension *O*. The latter distance is called the **height** of the pendulum and is represented in the figure by *h*.

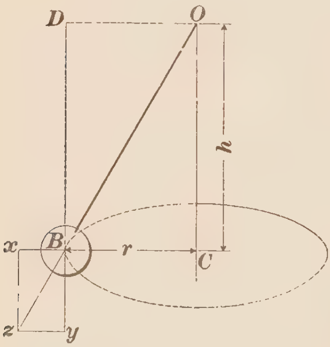

FIG. 1.

When the pendulum is revolving, the ball is acted on by three forces, namely: Gravity, which is equal to the weight of the ball and acts downwards; centrifugal force, which acts horizontally outwards; and the pull in the cord. These three forces may be respectively represented by the lines $B\,y$, $B\,x$, and $B\,z$. Considering the point of suspension O as a fulcrum, it is evident that the force represented by $B\,y$—the weight of the ball—tends to revolve the ball downwards with O as a center and a lever arm r equal to the perpendicular distance $D\,O$ from the line of action of the force to the fulcrum. Similarly, the centrifugal force—represented by the line $B\,x$—tends to revolve the ball in an upward direction around O with a lever arm equal to the height h. Since the force represented by $B\,z$ acts along a line passing through O, it has no tendency to rotate the ball around that center; it is, therefore, evident that the ball will be in equilibrium when the product of the centrifugal force multiplied by the distance h is just equal to the product of the weight of the ball multiplied by the distance r.

If the centrifugal force is increased by an increase in the speed of rotation, the ball moves outwards; as a result, the length of the lever arm r is increased, while h is diminished until the turning effects of the two forces are again equalized.

8. The Simple Pendulum Governor.—Fig. 2 shows a revolving pendulum as applied to a form of governor much used on Corliss engines. For the purpose of making the governor symmetrical and preventing the bending action on the spindle that would result from the effect of a single weight, two balls B, B are used. These balls are suspended from the collar C_1 that is fastened to and rotates with the spindle S.

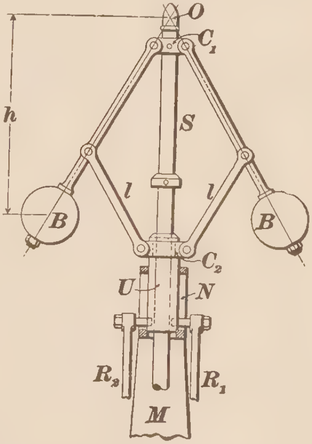

FIG. 2.

Links *l*, *l* connect the arms to which the balls are attached to a collar C_2 that turns freely on the sleeve *U*. This sleeve does not rotate, but is free to slide up and down on the spindle. When the balls fly outwards, the sleeve is pulled upwards by the links *l*, *l* and the collar C_2. This motion is transmitted to the valve gear by the connecting-rod R_1.

9. The Dashpot.—To prevent sudden fluctuations of the governor, a rod R_2 operates a piston in a cylinder filled with oil. A small opening, either through the piston or through a passage outside of the cylinder, connects the spaces above and below the piston in the cylinder. As the piston moves up or down in the cylinder, the oil flows from one side to the other through the small passage. By regulating the size of the opening, the rate of flow of the oil and, in consequence, the rate of motion of the piston can be regulated. This checking device is called a **dashpot.**

10. Defects of the Simple Pendulum Governor. From what has been said, it is evident that the weights of a centrifugal governor can change their position with respect to the axis only when there is a change in their speed of rotation; it is, consequently, evident that such a governor cannot keep the speed of an engine at an absolutely uniform rate under all loads. For example, consider an engine with a throttling governor; when the engine is running *light*—without any load—the governor valve will be opened just wide enough to admit the steam required to overcome the frictional resistances in the engine and to keep it running at a uniform speed. When the engine is *loaded*, the valve must be opened wider in order to admit the steam required to do the extra work. This variation in the opening of the valve can be accomplished only by a change in the position of the governor weights, and this change can take place only when the speed of the engine changes.

For some purposes, a considerable variation in speed is permissible, but in most cases it is desirable to keep the range between the speeds at no load and full load as small as is practicable; for example, in electric lighting, a total range in speed of 4 per cent. cannot be exceeded without causing a disagreeable flickering in the lights. For a given change in speed, the change in position of the weights of a simple pendulum governor is relatively very small, and the relative change decreases rapidly as the speed of rotation of the governor increases. For example, with a total variation in speed of 4 per cent., the theoretical variation in the height h (see Fig. 1) of a simple revolving pendulum is only .57 inch when the weight makes 50 revolutions per minute, and at 200 revolutions the variation in height is only .035 inch. From this it is evident that a governor of this type is impracticable for high speeds, and even at low speeds the range in motion for a permissible range in speed is so small that it is difficult to move the controlling valve through a distance great enough to give a satisfactory range in opening.

11. Another defect in the simple pendulum governor is the fact that *gravity*—the force that acts in opposition to the centrifugal force developed by the rotation of the weights—is too small to quickly overcome the inertia of the weights when there is a sudden reduction in the centrifugal force. This makes the action of the governor slow in responding to a sudden increase in load; the result is that before the governor can move far enough to open the valve and admit the steam required for the greater load, the speed of the engine will drop considerably below that corresponding to the load. Any attempt to make the governor act more promptly by increasing the weights will be futile, since the mass whose inertia must be overcome is increased in the same proportion as the increase in weight.

12. The **weighted-pendulum governor,** also called the **Porter governor** from the name of its inventor, is a modification of the simple pendulum governor that adds greatly to the range of motion of the centrifugal weights for a given range in speed. In this governor (see Fig. 3, which shows the form in which it is applied to the Porter-Allen engine) the centrifugal weights are small and their centrifugal force is made comparatively great by running them at a high speed. In addition to their own weight, the balls must lift a large weight, or counterpoise, *w* that is free to slide up and down on the spindle *r* and is lifted by means of the links *l*, *l*. The counterpoise *w* revolves with the spindle. At its lower end it is fitted with a collar that gives motion to the lever *o* that transmits the motion to the gearing that operates the valve. The lever *o* carries a weight *s* that can be adjusted along the lever so as to change the speed at which the engine will run by a small amount. The dashpot *d* checks any tendency of the governor to fluctuate too rapidly. The governor is driven by a belt from a pulley on the engine shaft to the pulley *p*, which drives the spindle *r* by means of a pair of bevel gears.

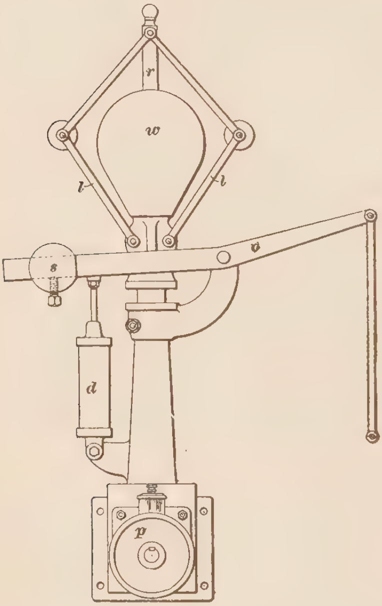

FIG. 3.

The effect of the counterpoise is to add to the resistance against which the centrifugal force developed by the rotation of the balls must act without changing the centrifugal force itself, as would be the case if extra weight were added to the balls. The result is that the balls must revolve at a higher speed in order to develop a centrifugal force great enough to overcome the added resistance. The change in the height of the governor for a given change in speed of rotation is also greatly increased. This is called **increasing the sensitiveness of the governor.**

SPRING-LOADED GOVERNORS.

13. Springs are often used instead of the counterpoise for increasing the resistance opposing the centrifugal force of the revolving weights of a pendulum governor. Fig. 4 shows a form of the **Gardner throttling governor** with a helical spring enclosed in a case provided with a hand wheel and screw *w* by means of which the amount of compression of the spring, and, in consequence, the resistance, can be varied. By changing the compression of the spring, the speed of the engine can be varied through a wide range. The spindle and flyballs are driven through the bevel gears by a belt on the pulley *p*. As the balls fly outwards, they force the spindle *t* downwards; this operates the lever *l* and compresses the spring, which thus acts to oppose the outward motion of the balls. The motion of the spindle is transmitted to the valve by means of the valve stem that is attached to the lower end of the spindle.

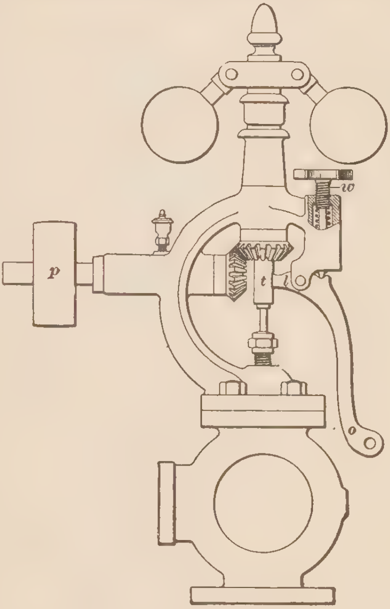

FIG. 4.

The extension *o* of the lever *l* provides a means of attaching a cord or other connection by means of which the speed of the engine can be controlled from a distance. This device, which is much used in connection with engines for driving sawmills, is called a **sawyer's lever.**

14. Fig. 5 shows the **Pickering governor** that has three flyballs attached to the flat springs *s*, *s*. In this governor the resisting force is furnished almost entirely by the resistance of the springs and is but little affected by the weight of the balls. As a means of varying the resistance and the speed of the engine, the governor is provided with a coil spring *O* wound around a spindle that carries the fork *e*. The tension of the spring presses this fork upwards against the governor spindle and thus increases the resistance in proportion to the tension that may be regulated by the worm-wheel and worm operated by the hand wheel *x*.

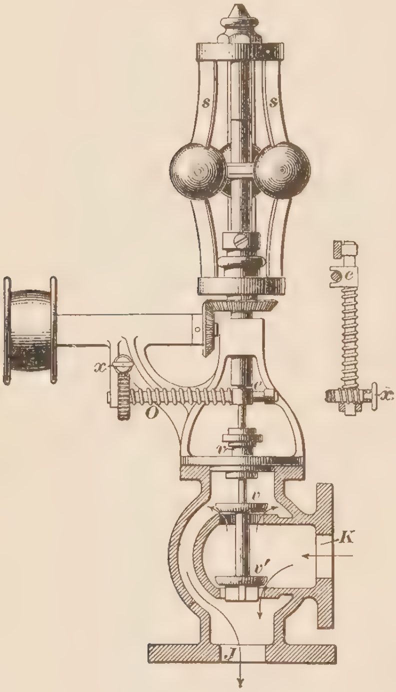

FIG. 5.

As the governor balls move outwards against the resistance of the springs, they lower the valves *v* and *v'* and thus partially shut off the steam supply. Steam enters at *K* and flows in the direction of the arrows through the opening *J* to the steam chest. The two valves *v* and *v'* are balanced; that is, the pressure on the top of one is balanced by a nearly equal pressure on the lower side of the other; this makes the resistance to their motion very small and makes it possible for the governor to move them easily.

15. Advantages of the Spring-Loaded Governor.—The spring of the spring-loaded governor serves nearly the same purpose as the counterpoise of the weighted-pendulum governor. The spring-loaded governor, therefore, shares with the weighted-pendulum governor the advantages of a high speed of rotation and a high degree of sensitiveness. With the spring-loaded governor, however, the force opposing the centrifugal effect of the flyballs can be made as great as is desired without the necessity of introducing a heavy weight whose inertia must be overcome by comparatively small forces before the governor can respond to a change in speed.

Further, after the weights of a governor have been set in motion, their inertia must be overcome before they can be brought to rest in the position corresponding to the change in speed; the consequence is that if a heavy weight, like the counterpoise of a weighted-pendulum governor, is used, its inertia tends to carry the governor beyond the correct position, and the valve is opened or closed more than enough to meet the change in load. This results in a change in speed that carries the governor in the opposite direction, and a series of fluctuations in speed and governor positions, known as **hunting** or **racing,** are set up. A dashpot tends to obviate this trouble by introducing a resistance that prevents the parts of the governor from attaining a high speed and consequently a large amount of energy to be overcome. Such a resistance, however, while it prevents violent fluctuations and hunting, makes the governor slower in adjusting itself to a change in load and so reduces the closeness of regulation that may be obtained unless a very heavy flywheel is used. With the comparatively light weights of a spring-loaded governor, the momentum of the moving parts is made small in proportion to the forces and resistances to be overcome; this type of governor, therefore, responds more quickly to changes in speed and is much less subject to a tendency to hunt. Another advantage of the spring-loaded governor is the facility with which the speed of the engine can be changed by changing the tension of the spring.

SHAFT GOVERNORS.

16. Governing by Shifting the Eccentric.—It has been previously stated that the work done in an engine cylinder can be regulated by varying either the point of cut-off or the point of compression, or by varying both of these points together. One of the most common methods of accomplishing this is by shifting the eccentric so as to vary its throw and angle of advance.

To illustrate the effect on the action of a slide valve of changing the position of the eccentric with reference to the crank and shaft, consider Figs. 6 to 11. In these figures

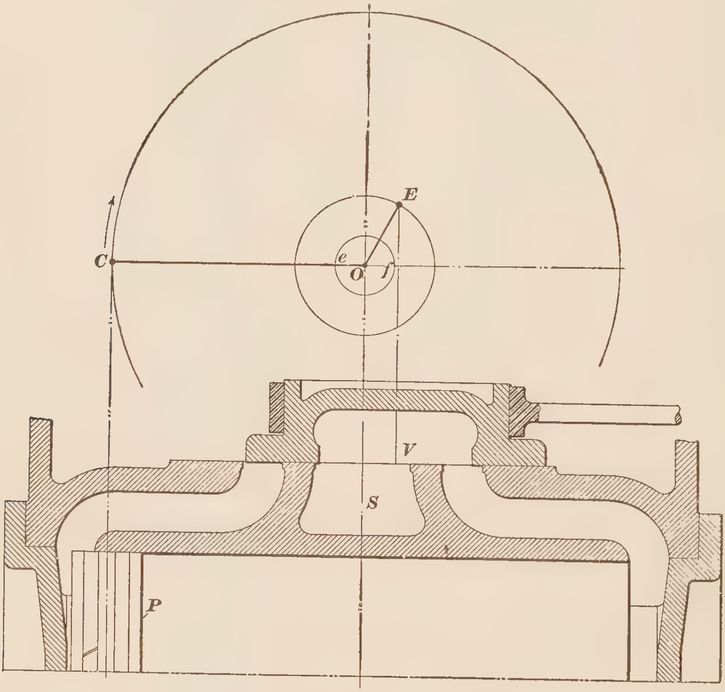

FIG. 6.

OC represents the crank, the motion of which is assumed to be in the direction of the arrow. The line OE represents the eccentric in the position of its greatest throw. Immediately below the circles representing the paths of crank and eccentric are shown part sections of a steam-engine cylinder with the piston P and slide valve V. In each figure the point O, representing the center of the shaft, lies in the line *O S* through the middle point of the valve seat, corresponding to the middle line of the valve when in its central position, and the piston and valve are in positions corresponding to the positions they would have if they were, respectively, connected to the crank and eccentric by rods so arranged that their angularity had no effect on the relative motions of crank and piston or eccentric and valve.

In each figure the distance of the valve from its central position is equal to the perpendicular distance between the line *O S* and the line *E V* or *E′ V* drawn parallel to *O S* through the eccentric center positions *E* or *E′*. The radii of the small circles *e f* are equal to the lap of the valve; consequently, when the lines *E V* or *E′ V* produced are tangent to this circle, as in Figs. 10 and 11, the distance of the valve from its central position is equal to the lap, and the edge of the valve coincides with the edge of the steam port. Further, in Figs. 6, 7, 8, and 9, the distance of the line *E V* or *E′ V* from the extremity *f* of the diameter *e f* is equal to the port opening for the corresponding eccentric and valve positions.

17. Effect on Lead by Shifting Eccentric Center Along a Line at Right Angles to Center Line of Crank. In Fig. 7 the crank is on the center, the piston is at the left-hand end of its stroke, just ready to begin its stroke to the right, and the valve has moved from its central position far enough to give the desired lead. Now, with the crank in the dead-center position, let the eccentric be shifted so that its center moves along a straight line at right angles to the center line *O C* of the crank. An inspection of Fig. 7 shows that this movement of the eccentric does not change the position of the valve; with the eccentric at any point, as *E′* or *E″*, in the perpendicular through *E* to the center line *O C*, the lead of the valve is the same as when the eccentric is in the extreme outer position *E*. When the eccentric center is in the position *E″*, coinciding with the line *O C*, the position in which the travel of the valve is least, the maximum port opening is equal to the lead. In no position of the eccentric is the valve prevented from opening the port at the beginning of the stroke; at the position E'', however, the amount

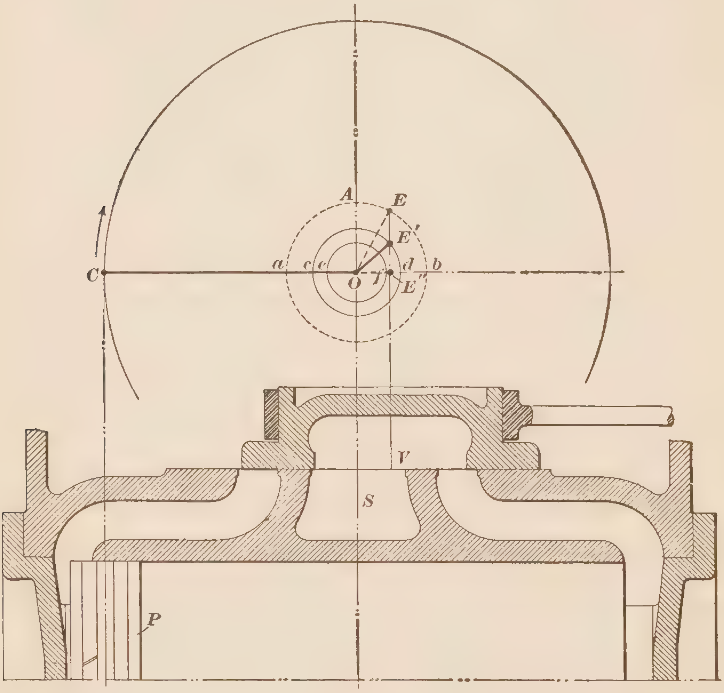

FIG. 7.

of opening is small and the period during which the valve remains open is short.

18. Effect on Valve Travel and Angle of Advance. Fig. 7 shows that the change of the position of the eccentric center from E to E' has shortened the eccentric radius, thus reducing the throw of the eccentric from the distance represented by the diameter of the circle $a\,b$ to the distance represented by the diameter $c\,d$. Another effect of the change in position has been to increase the angle of advance from the value $A\,O\,E$ to $A\,O\,E'$.

19. Effect on Maximum Displacement of Valve.—With the eccentric in its extreme outer position, the maximum displacement of the valve is equal to the radius OE. In its extreme position, the valve opens the left port fully, as is shown in Fig. 8. Full opening occurs when the piston has moved through about one-quarter of its stroke from the left towards the right. The maximum displacement of the valve when the eccentric center has moved to the position E' (see Fig. 9) is equal to the radius OE'. In this case, the port is opened only a part of its full width. Owing to the increase in the angle of advance, the maximum port opening occurs when the piston has moved through only one-eighth of its stroke from the left towards the right.

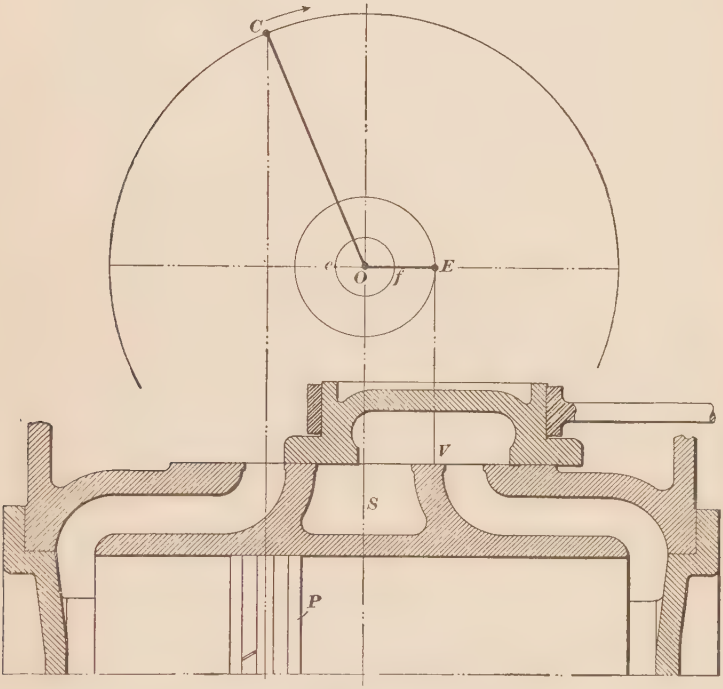

FIG. 8.

20. Effect on Point of Cut-Off.—In Figs. 10 and 11 the valve is in the cut-off position that is, it has moved to the left until it has just closed the port to the admission of steam. In Fig. 10 the eccentric is in its extreme outer position and the piston has traveled more than three-quarters of its stroke. When, however, the eccentric center has shifted to the position E' (see Fig. 11) the valve closes the port and cuts off steam before the piston reaches the middle point of its stroke.

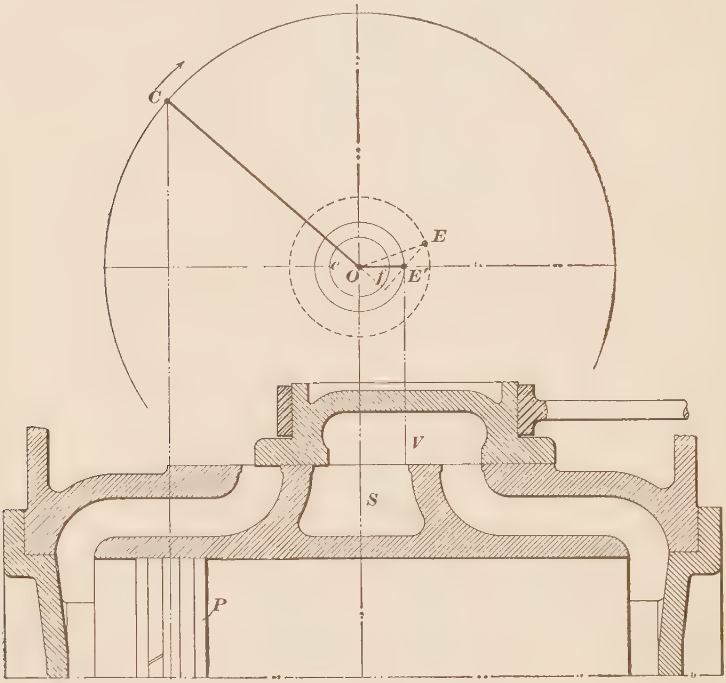

FIG. 9.

21. Effect on Release and Compression.—By a construction similar to Figs. 10 and 11, it is easily shown that the effect of shifting the eccentric from its outer position E along the line $E E'$ is to hasten release and compression. Thus, considering either end of the cylinder, as the eccentric center moving from *E* in the direction of *E'* approaches nearer the center line *O C* of the crank, release takes place earlier in the stroke, as the piston moves from

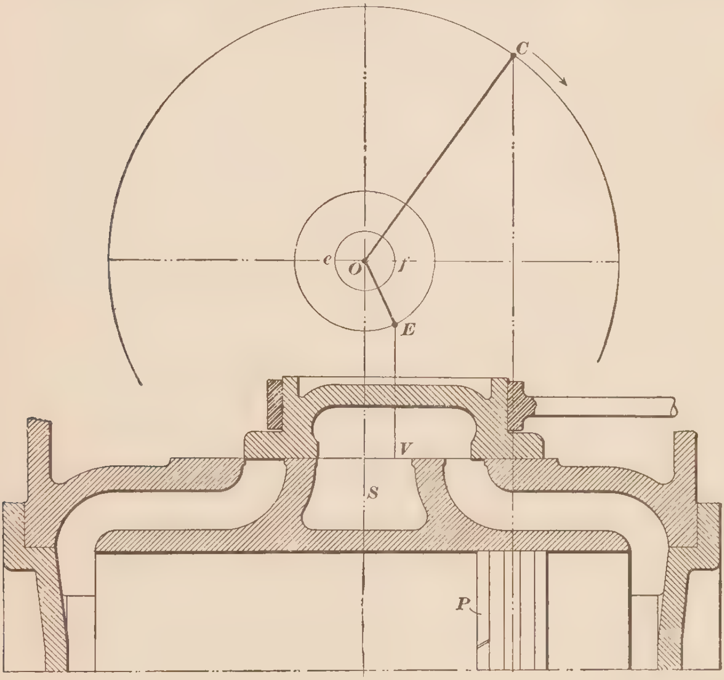

FIG. 10

that end and the port is closed so as to prevent the escape of exhaust steam earlier in the return stroke. Compression begins earlier and the resistance to the motion of the piston is thus increased.

22. General Review of Effects of Shifting the Eccentric.—A general consideration of the effects observed in connection with the diagrams, Figs. 6 to 11, shows that the combined effect of a reduction in the eccentric radius and an increase in the angle of advance is to reduce the valve travel and port opening and to make the events of maximum port opening, cut-off, release, and compression occur earlier in the stroke. These diagrams also show that if the eccentric is shifted in such a manner that its center

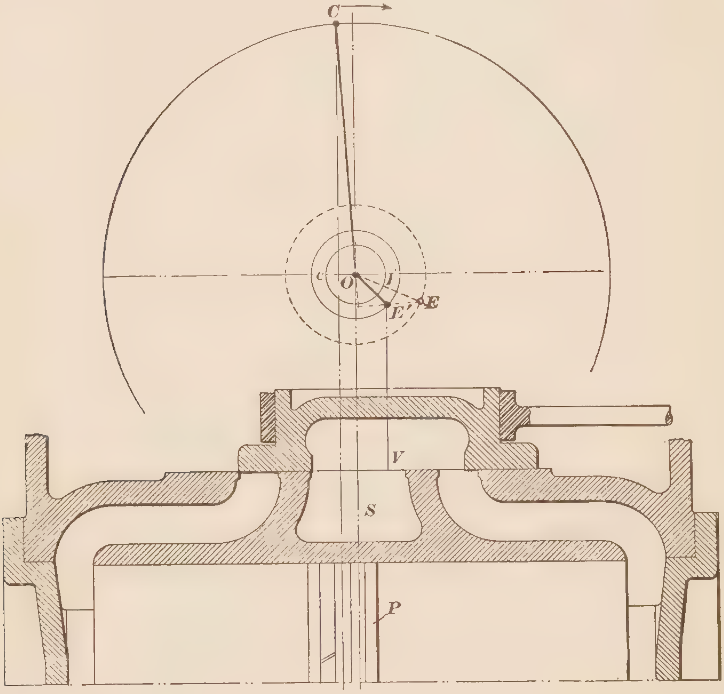

FIG. 11.

moves in a line perpendicular to the center line of the crank, the lead of the valve will not be changed.

23. In the **Erie engine governor** (see Fig. 12) we have one of the early types of shaft governors in which the eccentric is shifted in practically the same manner as is illustrated by the diagrams, Figs. 6 to 11. A frame *A B* is keyed to the shaft *O*. Two bell-crank levers *L*, *L'* are pivoted to this frame at *a* and *a'*. At one end these levers are enlarged so as to form the weights *W*, *W'*; at the other end they are attached to the links *l*, *l'* that connect with the eccentric *E* The eccentric is slotted, as shown, so that it may be shifted in such a manner that its center *c* will move in a line perpendicular to the center line *O C* of the crank. As the shaft

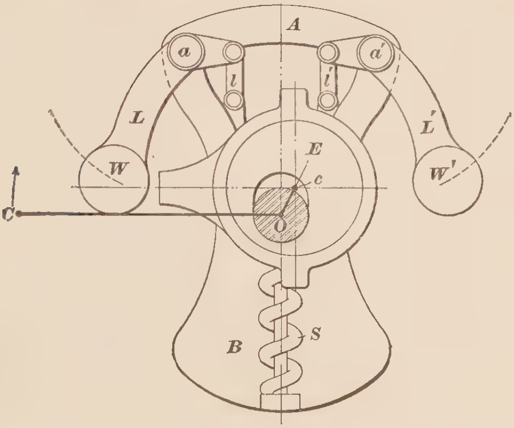

FIG. 12.

with the frame and governor revolves, the centrifugal force of the weights *W*, *W'* tends to cause them to fly outwards along the dotted arcs and so force the center *c* of the eccentric nearer the center line *O C* of the crank. This motion is resisted by the action of the spring *S*, which is of such a length that when put in place it is compressed and exerts a pressure that tends to hold the eccentric in its extreme outer position. This pressure, called the **initial tension** of the spring, resists the centrifugal force of the weights and prevents their moving the eccentric until the engine has reached the desired speed.

When the desired speed begins to be exceeded, the centrifugal force of the weights exceeds the resistance of the spring; they then move outwards along the dotted arcs, compress the spring, and shift the center of the eccentric nearer the line *O C*, thus shortening the stroke of the valve and increasing the angle of advance so as to make the cut-off, release, and compression take place earlier in the stroke. In this way, the work done in the engine cylinder is reduced to such an amount that a further increase in speed is prevented. If the load on the engine increases and the speed begins to decrease, the centrifugal force of the weights is reduced until the force of the spring is sufficient to shift the eccentric in the opposite direction until enough steam is admitted to the cylinder to do the work. In this way the speed of the engine is kept nearly uniform.

24. Shifting the Eccentric Center Along a Curved Path.—In order to give the eccentric center a straight-line path, the direction of motion of the eccentric must be controlled either by guides or by a complicated system of links. The latter method is impracticable and the former is not thoroughly satisfactory for the reason that the guides cannot be easily lubricated, and the wear and friction are, consequently, so great as to seriously interfere with the governor's action.

It is also generally desirable to arrange the motion of the eccentric in such a way that the lead of the valve will be as near its maximum as is practicable at early cut-off positions; at the same time, the lead should be zero when the governor is in its extreme outer position and the stroke of the valve is least; that is, with the governor in this position the valve should not open the ports at all, otherwise the engine would be in danger of "running away" when not loaded. We have seen that shifting the eccentric center along a straight line perpendicular to the center line of the crank does not vary the lead. Thus, referring to Fig. 7, it is seen that the shortest throw of the eccentric occurs when the center is in the position E'' on the center line $O C$. In this position, the lead of the valve is the same as it was when the eccentric center was in its extreme outer position E. On account of these difficulties, the usual method of guiding the motion of the eccentric is to attach it to an arm pivoted either to a disk or to the arm of a wheel keyed to the shaft. The center of the eccentric then moves in an arc whose center is the center of the pin to which the arm is pivoted.

25. Effect on Lead of the Point of Suspension of the Eccentric Arm.—Figs. 13 to 15 are diagrams in which are shown the effect on the lead of varying the position of the pivot of the arm to which the eccentric is attached. In these diagrams, *o* represents the center of the shaft, *o C* the crank; *E*, *E*′, etc., the eccentric center in various positions; and *P* the pivot point of the arm to which the eccentric is attached. As in Figs. 6 to 11, the radius *o f* of the circle *e f* is equal to the lap of the valve. In order to show more clearly the effect on the lead of a curved path for the eccentric center and a change in the point of suspension *P*, the eccentric radius *oE* and the change in eccentric position from maximum to minimum throw are shown much greater in proportion to the length *PE* of the arm than is usual in practice.

In Fig. 13 is shown a common method of suspension, in which the pivot *P* is located on the center line *o C* of the crank. With the eccentric in its extreme outer position *E* the lead is least, being represented in the diagram by the distance *f a*. As the eccentric center shifts towards the earliest cut-off position *E*″, the lead increases quite rapidly at first, as is seen by the distance *f b*, which represents the lead when the eccentric center is in the position *E*′. With the pivot in this position, the lead is greatest when the eccentric center is in its point of earliest cut-off *E*″. The increase in lead as the cut-off becomes earlier is an advantage of this method of suspension, since at the beginning of the stroke it gives the valve a liberal opening and secures a full pressure of steam in the cylinder, thus neutralizing in some degree the tendency of the short travel of the valve to produce wiredrawing of the steam. A disadvantage is that in case the load is suddenly taken

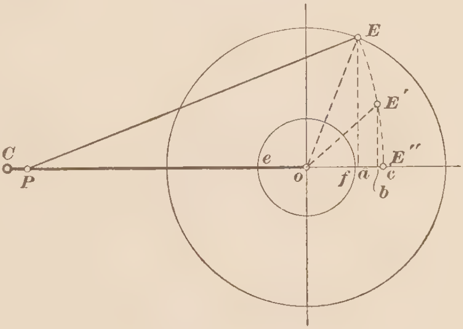

FIG. 13.

off the engine, it may attain a dangerous speed, owing to the comparatively large port opening at the position of least throw of the eccentric.

By putting the pivot on the center line of the crank, but on the side of the shaft opposite the crankpin, as in Fig. 14,

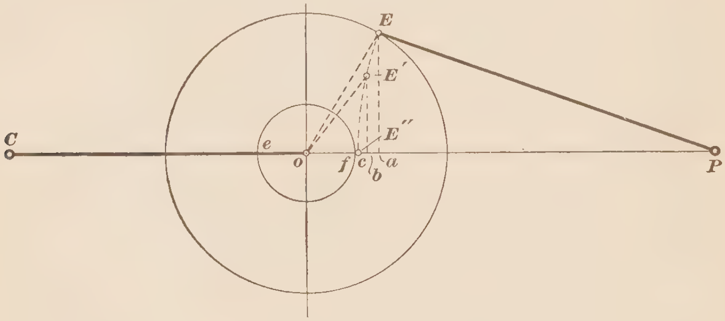

FIG. 14.

the lead decreases as the eccentric center nears the center line of the crank, and can readily be made zero when the eccentric center is in its shortest throw position. This method, however, has the disadvantage of wiredrawing the steam at early cut-offs, with which disadvantage, however, is combined the advantage that in case all load should suddenly be thrown off the engine, the restricted port opening due to the eccentric being shifted to its position of least throw will tend to prevent the engine from attaining a dangerous speed.

Attention is called to the fact that Figs. 13 and 14 are drawn for a direct valve and direct rocker-arm or an indirect valve with a reversing rocker-arm. For an indirect valve with a direct rocker-arm or a direct valve with a reversing rocker-arm, the point of suspension *P* of the eccentric link must lie on the side of the shaft opposite the crankpin, in order that the lead may increase at early cut-offs, and on the same side of the crankpin, in order that the lead may decrease at early cut-offs.

26. In Fig. 15 we have a method of suspension that gives more satisfactory results than either of the others, possessing as it does most of the advantages of both. The pivot is located on the same side of the shaft as the crank, but instead of being placed on the center line, it is located at one side. The change in lead is very slow during the motion of the eccentric center through the range corresponding to the usual load. When, however, the eccentric center approaches the position of minimum throw, the lead is rapidly reduced and can readily be made zero at the limit.

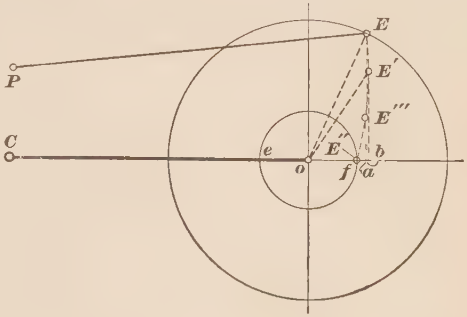

FIG. 15.

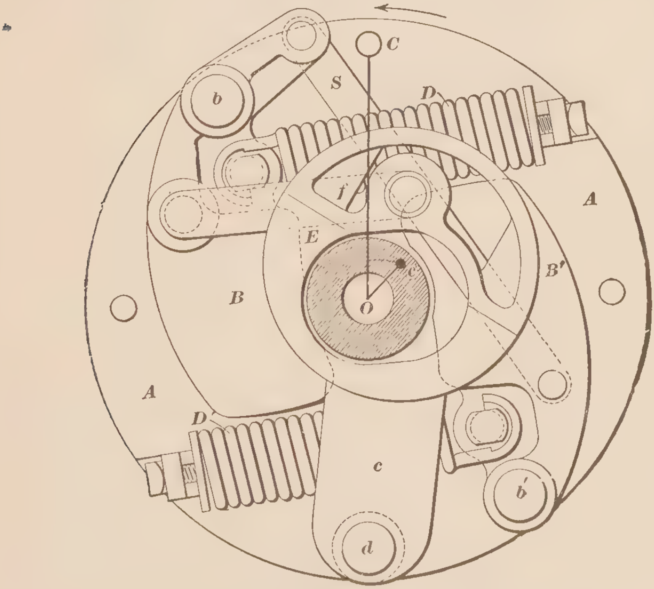

FIG. 16.

27. The **Westinghouse governor** (see Figs. 16 and 17) illustrates the method shown in Fig. 13. A disk *A* is attached to the shaft and the weights *B* and *B′* are pivoted to this disk at *b* and *b′*. The eccentric *E* is attached rigidly to the arm *c*, which is pivoted to the disk at *d*. The two weights are connected by a long link *S* in such a way that they always move together. The short link *f* connects the weight *B* with the eccentric; any motion of the weights is thus transmitted to the eccentric so as to change the position of its center with respect to the center line *O C* of the crank.

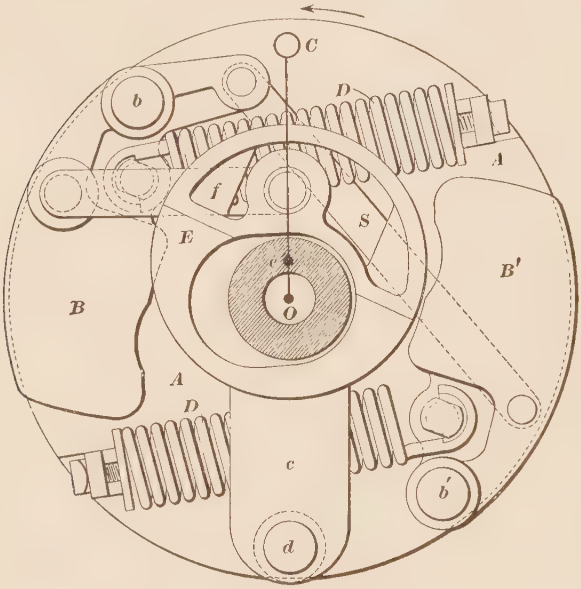

Fig. 17.

Fig. 16 shows the weights in their inner position, while the eccentric is in the position of its greatest throw. As the shaft revolves, the centrifugal force of the weights is resisted by the tension of the springs *D*, *D′*. When the speed becomes great enough to enable the centrifugal force to overcome the resistance of the springs, the weights move outwards and shift the eccentric about its pivot *d* until the travel of the valve is adjusted to the work to be done. Fig. 17 shows the governor when the weights are in their extreme outer position. The eccentric center then takes the position e', nearly on the center line $O\ C$ of the crank, having moved from the position e of Fig. 16 along the dotted arc; this is the position of earliest cut-off.

It will be noticed that in this governor the eccentric lies behind the crank at an angle of 90° minus the angle of advance. The reason for setting the eccentric in this position is that an indirect valve is used with this governor. If a direct valve were used, the crankpin would be on the opposite side of the shaft, which would make the relative positions of crank and eccentric agree with the positions shown in Fig. 13.

INERTIA GOVERNORS.

28. Effect of Inertia of Weights on the Action of a Governor.—In the types of governors so far considered, the force that acts to shift the valve or eccentric is principally the difference between the centrifugal force of the revolving weights and the resistance opposed to this force by gravity or the governor springs. In the comparison of weighted- and spring-loaded pendulum governors (see Art. **15**), attention was called to the resistance that the inertia of the moving parts of a governor offers to a change in their position, and it was pointed out that the effect of inertia is first to oppose the motion of the weights, then, after they have been set in motion, to carry them beyond the point at which there is equilibrium between their centrifugal force and the resistance. Inertia, thus, has the effect of a disturbing force; it makes the governor slow to respond to the action of the forces induced by a change in the speed of rotation, and thus permits a considerable change in speed before the governor adjusts itself to the change in load; it also acts to carry the weights beyond the point corresponding to the correct valve opening or eccentric position, and so sets up a series of vibrations and the governor "hunts" or "races."

29. In the spring-loaded pendulum governor, inertia is made small by the use of small weights, while the useful forces are made comparatively large by running the governor at a high speed and using powerful springs; in this way the disturbing effect of inertia is kept from becoming serious. In the shaft governor, however, the speed of rotation is limited to the speed at which the engine may run; it is, therefore, impossible to obtain enough centrifugal force to properly control the valve without the use of heavy weights, and this, in turn, means a corresponding increase in inertia.

30. In the **inertia governor,** the effect of inertia instead of acting as a disturbing force is made to assist the weights in adjusting themselves to changes in speed; such a governor can, therefore, be made to respond very promptly to the slightest variations in load and at the same time be free from a tendency to hunt. Fig. 18 shows the type of inertia governor used on the McEwen engine and will serve to illustrate the principles involved in all governors of this

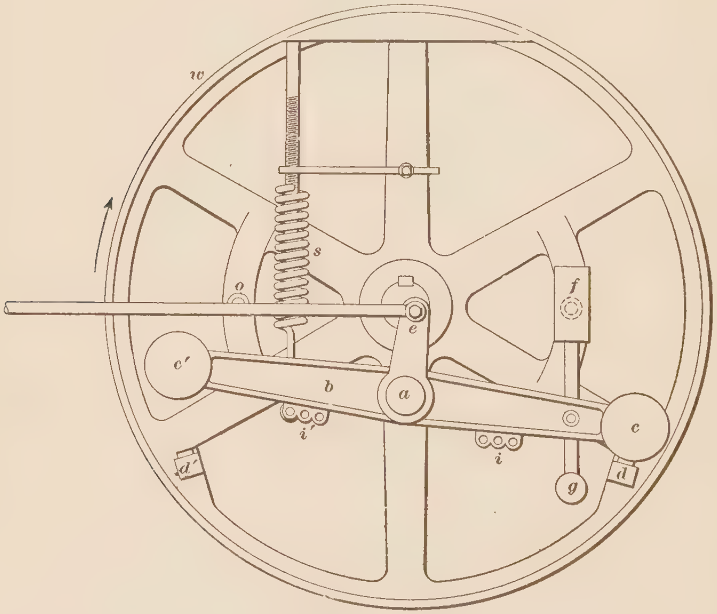

FIG. 18.

class. The parts are shown in the positions occupied when the engine is standing still. The wheel *w*, sometimes called the governor case, is keyed to the engine shaft in the same manner as is an ordinary flywheel. Pivoted to this wheel, by a roller bearing on the pin *a*, is the inertia weight or bar *b*, the two ends *c*, *c′* of which are hollow; this bar is symmetrical with respect to the center of the pin *a*. The spring *s* is attached at one end to the rim of the wheel and at the other to the bar *b*. The end *c′* of the bar *b* is weighted by being partly filled with shot, and as the wheel revolves the centrifugal force of this weight tends to carry *c′* outwards; this tendency is resisted by the pull of the spring *s*, the initial tension of which is sufficient to overcome the centrifugal force until the required speed is reached. When the speed begins to exceed that desired, the centrifugal force of the weight in *c′* throws out that end of the bar; that motion shifts the eccentric pin *e*, which serves the purpose of an eccentric, nearer the center of the shaft, thus shortening the throw of the eccentric and increasing the angle of advance in practically the same way as is illustrated in Figs. 6 to 11. Up to this point the action of the governor is the same as that of the centrifugal governors previously discussed.

31. Now let us consider the effect of a sudden decrease of the load on the engine: As the load, and, consequently, the resistance to the turning of the shaft suddenly drops, the speed of rotation of the shaft and governor wheel quickly begins to increase. The only way in which this increase in the speed of rotation can be imparted to the heavy bar *b* is through the action of the flexible spring *s*. Owing to the weight and length of the bar, its inertia offers considerable resistance to a sudden change in its angular velocity; the result is that the wheel advances faster than the bar, which has the effect of shifting the eccentric pin *e* nearer the center of the shaft and causing the cut-off to take place earlier, thus preventing a serious increase in speed before the centrifugal force can act. If the load on the engine is suddenly increased, the speed of the wheel is quickly checked; the resistance of the bar to a sudden change in its speed of rotation causes it to move ahead of the wheel and thus shift the eccentric pin so as to make the cut-off take place later; in this way enough steam is admitted to the cylinder to meet the demand. In case of a change in the speed of the shaft, the inertia of the bar acts at once to shift the eccentric in the required direction, and so prevents the change from becoming serious before the centrifugal force can operate. As soon as the eccentric reaches the position corresponding to the work to be done, the speed of the engine is controlled, as before, by the relation between the centrifugal force of the weight in c' and the tension of the spring.

The stops d' and d limit the range of motion of the bar. A dashpot f, consisting of a cylinder in which a loose fitting piston works, checks the motion of the bar and prevents too rapid changes in the eccentric position. The centrifugal force of the dashpot piston is counterbalanced by that of the weight g on the other end of the piston rod; this prevents the piston from bearing heavily against the walls of its cylinder and so reduces friction and wear. The lug i', to which the spring is attached, has three holes, by means of which the lever arm at which the pull of the spring acts can be changed. A similar lug i provides a means of attaching the spring to the opposite end of the bar, and the dashpot may likewise be shifted to the lug o. In this way the direction of rotation may readily be reversed.

32. Path of Eccentric Center With Inertia Governors.—With the governor shown in Fig. 18, the eccentric is attached directly to the inertia bar, and the point of suspension of the bar is on the center line of the crank and on the side of the shaft opposite that of the crankpin. A governor of this type has the advantage of having the fewest possible number of parts with only one joint to be fitted up and kept in order; as is shown in Fig. 14, however, the position of the point of suspension with a direct valve and rocker-arm, or an indirect valve and reversing rocker-arm, results in an effect on the lead as the cut-off becomes earlier that has some disadvantages.

In order to use an inertia governor with an eccentric suspended as in the diagram Fig. 13, it is necessary to connect the eccentric to an arm separate from the inertia bar and to connect this arm to the inertia bar in such a manner that its direction of rotation, with respect to the governor case,

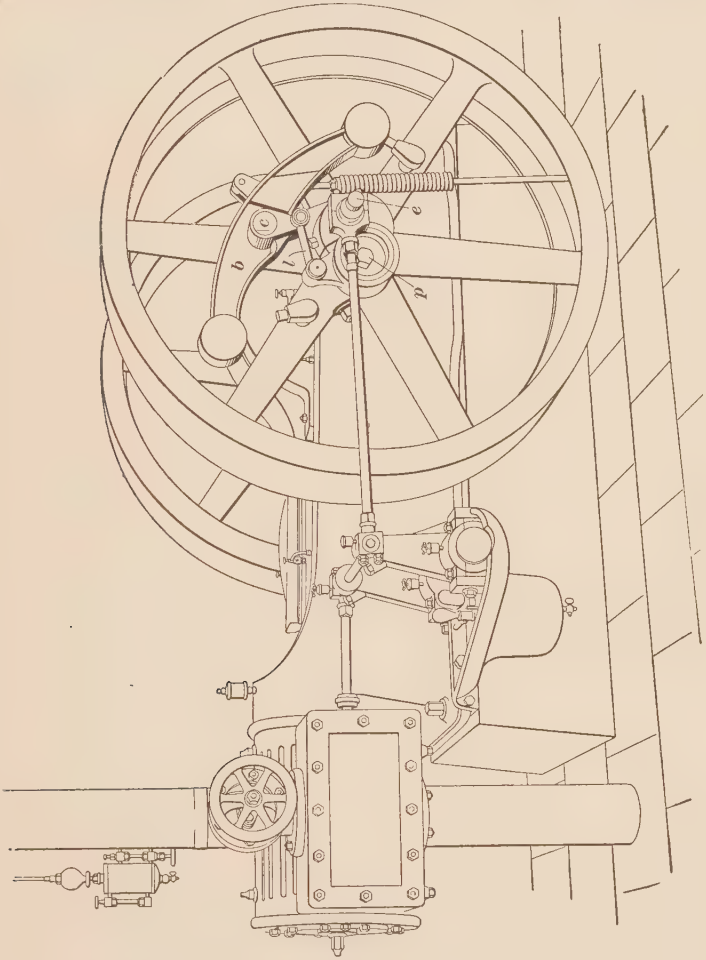

Fig. 19.

will be opposite to that of the inertia bar. Fig. 19 shows how this principle is applied to the governor of the American Ball engine. The eccentric pin *e* is attached to an arm that is pivoted at *p*. The inertia bar *b* is pivoted at *c* and is connected to the eccentric arm by the link *l*. An inspection of the figure shows that the direction of rotation of the eccentric arm around the pivot *p* is opposite that of the inertia bar around its pivot *c*.

ISOCHRONISM.

33. Condition of Isochronism.—The formula for the centrifugal force of a weight revolving around an axis outside its center of gravity is

$$F = .00034\ W R n^2,$$

where F = centrifugal force in pounds;
W = weight in pounds;
R = radius of the circle described by the center of gravity of the weight, in feet;
n = number of revolutions per minute.

This formula shows that the centrifugal force of a governor's weight increases at the same rate as the distance R from the center of gravity of the weight to the axis. From this it is evident that if a governor is so made that the resistance to the outward motion increases at the same rate as the increase in the radius of the circle in which the weight rotates, there will be a speed at which there will be equilibrium between the centrifugal force and the resistance for any position of the weight. It also shows that if there is no change in the resistance, any increase in speed above that at which the resistance balances the centrifugal force, the latter force will carry the weights to their extreme outer position, while the weights will be held in their extreme inner position for any speed below that at which there is equilibrium between the forces. This condition of constant ratio between the centrifugal force and the resistance opposed to it is called **isochronism,** and a governor proportioned in such a manner that this condition is fulfilled is called an **isochronous governor.**

34. Limitations to Isochronism.—Isochronism is desirable in that it implies a sensitive governor that will respond to the slightest changes in speed. A perfectly isochronous governor, however, will be too sensitive for any practical use. At the one speed at which there is equilibrium, the weights will stay in any position and will have no control over the steam supply. The slightest change from this speed will have the effect of shifting the weights to one or the other *extreme* position, and either shut off the steam supply entirely or open the admission valve wide. This will cause rapid fluctuations of the governor from one extreme position to another; the distribution of steam will be most unsatisfactory, and although the variation in speed might be small, the speed of the engine will be constantly changing.

The friction of the joints and the inertia of the moving parts of a governor disturb the relation between the forces and make perfect isochronism impossible. Unfortunately, however, these resistances do not assist materially in steadying the governor and preventing violent fluctuations; in fact, they often add to the violence of the fluctuations by resisting the action of the forces until the change in speed is great enough to overcome the added resistance with which they oppose the shifting of the weights in either direction.

35. Isochronous Pendulum Governors.—Since, for a given speed, the balls of a simple pendulum governor can have but one position in which there is equilibrium between their centrifugal force and the effect of gravity, such a governor is not isochronous. By suspending the balls from crossed arms, however, as in Fig. 20, the height of the governor, i. e., the vertical distance between the centers of the balls and the point where the center lines of their arms intersect the axis of the spindle, remains nearly constant for any position of the balls, and the condition of isochronism is very nearly fulfilled. By a careful choice of the relation between the length of the arms and the distance of their points of suspension from the axis, the crossed-arm governor can be made very sensitive. A number of governors working on this principle have been built, but they have the defect common to all gravity governors of requiring heavy weights in order to be sufficiently powerful, and, in consequence, are subject to the disturbing effects of inertia.

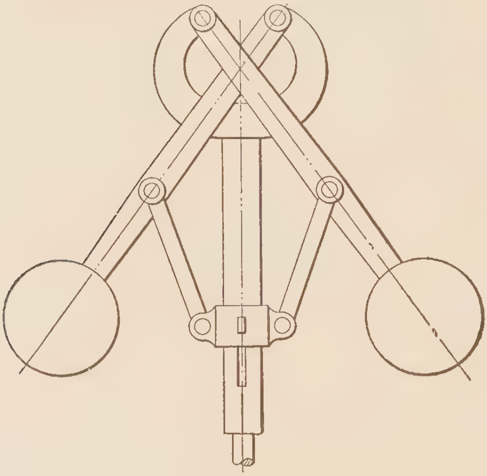

FIG. 20.

36. Isochronism With Spring-Loaded Governors.—A spring-loaded governor of either the revolving-pendulum or the shaft-governor type can be made practically isochronous by so adjusting the spring that, as the weights fly outwards, the resistance of the spring increases at such a rate as to just balance the increase in centrifugal force. Since a perfectly isochronous governor would be too sensitive for practical use, it is usual to so adjust the spring that its resistance will increase somewhat faster than the increase in the centrifugal force. With such an adjustment, as the speed of the engine increases the weights move outwards until they reach a position where the supply of steam meets the demand; in this position there will be equilibrium between the centrifugal force and the resistance until there is a change in speed. Any change in the load or the steam pressure causes a slight change in speed that shifts the weights to a new position. If the adjustment of the spring is correct and the governor is in good condition, the change in speed will be small. If the spring is adjusted to nearly the condition of isochronism, a change in speed will move the weights too far and the governor will hunt. If, on the contrary, the resistance of the spring increases at too great a rate, the change in speed required to shift the weights will be excessive and the regulation will be unsatisfactory.

37. Effects of Friction.—Friction acts as a force that resists the motion of the weights in either direction; in consequence, they will not immediately respond to a change in speed that destroys the equilibrium between the centrifugal force and its resistance; the speed will change until the difference is great enough to overcome the frictional resistance and, since friction is generally less between surfaces in motion than between surfaces at rest with respect to each other, the weights are likely to be shifted too far, thus setting up hunting. These considerations show the importance of governor pivots in which the frictional resistances are small, and of keeping all parts as well lubricated and free moving as possible.

38. Effect of the Dashpot.—A dashpot has a steadying effect on the action of a governor. Its resistance is very small when the motion is slow; it, therefore, permits the weights to respond promptly to a change in speed. If, however, the motion tends to become rapid, the resistance to the flow of the fluid through the small opening connecting the two ends of the dashpot cylinder acts as an opposing force that prevents the motion from becoming so rapid as to enable the inertia of the weights to carry them too far. The effect of the dashpot may be regulated by varying the size of the opening connecting the two ends of the cylinder.

ADJUSTMENT OF GOVERNORS.

ADJUSTMENT OF SIMPLE PENDULUM GOVERNORS.

39. A simple pendulum governor has a very limited range of speed through which it will have a satisfactory control of the valve, and no change in the mass of the weights or in their position on the arms has any effect on the height of the governor, i. e., the position in which it holds the valve for a given speed; owing to these facts, the only method of changing the speed of an engine controlled by this type of governor is to change the ratio between the diameters of the governor pulley and the pulley on the engine shaft. If the speed at which the governor should run and the diameter of either the governor pulley or the pulley on the engine shaft are known, the diameter of the other pulley for a given speed of the engine may be calculated by an application of rule **1**.

40. It is often required to know what change to make in the diameter of the governor pulley in order to secure a certain change in the speed of the engine. In such a case, the following rule may be used:

Rule 1.—*Knowing the speed at which an engine runs with a given diameter of governor pulley, to find the diameter of pulley for a required speed, multiply the given diameter by the required speed and divide the product by the known speed.*

Let S_1 = known speed of engine;
S_2 = speed of engine with new pulley;
D_1 = diameter of old governor pulley;
D_2 = diameter of new governor pulley.

Then,
$$D_2 = \frac{D_1 S_2}{S_1}.$$

Example.—With a governor pulley 6 inches in diameter, an engine runs at a speed of 175 revolutions per minute; what must be the diameter of pulley if the engine is to run at a speed of 190 revolutions per minute?

Solution.—Applying rule **1**, we have

$$D_2 = \frac{6 \times 190}{175} = 6.514 \text{ in.} = 6\tfrac{33}{64} \text{ in., nearly. Ans.}$$

WEIGHTED-PENDULUM GOVERNORS.

41. Weighted-pendulum governors are designed to run at a certain speed, and will work best when run at that speed. If they are provided with an adjustable weight, like the weight *s* of the Porter-Allen governor (see Fig. 3), the speed can be varied through a small range either by shifting the weight along the lever or by increasing or diminishing the weight. Shifting the weight nearer the end of the lever or increasing the weight adds to the resistance opposing the action of centrifugal force and thus increases the speed at which the governor must run for a given height. Shifting the weight in the opposite direction or making it lighter decreases the speed.

If it is desired to make a considerable change in the speed of the engine with a weighted-pendulum governor, it is best to change the size of the governor pulley in accordance with rule **1**.

ADJUSTMENT OF SPRING-LOADED PENDULUM GOVERNORS.

42. On account of the facility with which the resistance to centrifugal force can be varied by varying the tension of the spring, a considerable range in speed may be obtained with most spring-loaded pendulum governors without the necessity of changing the size of the governor pulley. Changes in the tension of the spring, however, have an effect on the sensitiveness of the governor; with a certain tension and speed, the relation between the centrifugal force of the weights and the resistance offered by the spring will give the most satisfactory results obtainable, and any variation from this will tend either to reduce the sensitiveness or to make the sensitiveness so great that the governor becomes unstable. For this reason, it is always desirable to run the governor as nearly the speed for which it was designed as is practicable and to use a combination of pulleys that will secure this speed. If it is desired to make a considerable permanent change in the speed of the engine, such change should be made by changing the size of governor pulley in accordance with rule **1**, and not by changing the tension of the spring.

ADJUSTMENTS OF SHAFT GOVERNORS.

43. Relations Between Spring Adjustment and Sensitiveness.—In Fig. 21 a weight W slides freely on a rod b attached to the rim of the wheel in such a position that its center line intersects the center of the shaft to which the wheel is fastened. The helical spring S opposes the outward motion of the weight W. If the wheel is rotated when the center of gravity of the weight is at the distance R from the center of the shaft and the spring is subjected to any initial tension T, the spring will prevent the weight from moving farther outwards along the rod until the number N of revolutions per minute is great enough to develop a centrifugal force F that is greater than T.

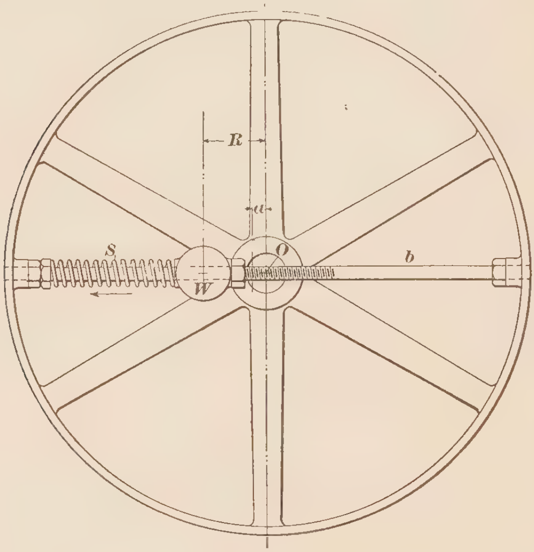

FIG. 21.

Now let the length of this spring be such that when the center of gravity of the weight W coincides with the center line of the shaft, the spring will just touch the weight without being compressed or extended and will, therefore, exert no pressure on the weight. It is a property of most springs that the force exerted by them varies directly with the distance through which they are deflected. If, therefore, the weight W is forced from the position where its center of gravity coincides with the center of the shaft and the position in which the spring S just touches it without being compressed or extended, to the position shown in the figure, the spring will be compressed through the distance R and it will resist the outward motion of the weight with a force directly proportional to that distance.

44. The formula for centrifugal force, $F = .00034\, W R n^2$, shows that the centrifugal force that will be developed by the weight when the wheel rotates around the center O is proportional to the distance R. The conditions are such that the compression of the spring for any position of the weight is proportional to the distance of the center of gravity of the weight from the center of the shaft; the centrifugal force increases as the weight flies outwards at the same rate as the increase in the force exerted by the spring and the condition of isochronism is fulfilled. Consequently, with this construction and adjustment of the spring, any speed of rotation that will enable the centrifugal force to overcome the resistance of the spring will cause the ball to move outwards through its whole range of motion.

Now, instead of touching the weight when its center of gravity coincides with the center of the shaft, let the spring be made a little shorter, so that compression will not begin until the center of gravity of the weight has been moved a distance a from the center line of the shaft. Under these conditions, when the weight is moved to the position shown in the figure the force exerted by the spring will not be proportional to the distance R, but to the distance through which the spring was compressed, that is, to $R - a$. If we denote by T the tension of the spring when the weight is at the distance R from the center, the magnitude of the tension will be expressed by the equation $T = K(R - a)$, in which K is a constant value depending on the dimensions of the spring and $(R - a)$ is the distance through which the spring is deflected. When the weight has moved outwards any distance r from the position shown, so that the distance from the center O is $R+r$, the spring will be deflected through a distance $(R+r-a)$, and the new tension T' will be $T'=K(R+r-a)$. The increase in tension due to the increase in the amount of deflection is $T'-T=K(R+r-a)-K(R-a)=Kr$, and the rate of increase is $\frac{T'-T}{T}=\frac{Kr}{K(R-a)}=\frac{r}{R-a}$. (*a*)

45. The centrifugal force for a given number n of revolutions per minute when the weight is at the distance R is $F=.00034\,WRn^2$. and when the weight has moved until its distance from the center is $R+r$, the centrifugal force for the same number of revolutions is $F'=.00034\,W(R+r)n^2$. The increase in centrifugal force due to the increase in the distance from the center is $F'-F=.00034\,W(R+r)n^2-.00034\,WRn^2=.00034\,Wn^2(R+r-R)=.00034\,Wn^2r$, and the rate of increase is $\frac{F'-F}{F}=\frac{.00034\,Wn^2r}{.00034\,Wn^2R}=\frac{r}{R}$. (*b*)

46. By comparing the rate of increase represented by formula (*a*) with that represented by formula (*b*), we see that the numerator is the same in each, but that the denominator in (*b*) is greater than that in (*a*) by the value a. Since the value of a fraction decreases as the value of its denominator increases, we see that the rate of increase in the tension of the spring is greater than the rate of increase in the centrifugal force. In consequence of the greater rate of increase in the tension of the spring, it follows that the number of revolutions required to enable the centrifugal force of the weight to balance the tension of the spring will be greater as the weight moves farther from the center. It is also evident that the increase will be greater as the distance a is increased.

By a similar course of reasoning, it can be shown that if the spring is lengthened so that when the weight is at a given distance R from the center the spring is compressed through a distance greater than R the centrifugal force will increase faster than the increase in the tension of the spring. With such an adjustment of the spring, when the weight was in its outer position there would be equilibrium between the force of the spring and the centrifugal force of the weight at a smaller number of revolutions per minute than when the weight was near the center. An engine with a governor spring so adjusted would run faster under a heavy load that required a large valve opening with the governor weights near their inner position than it would under a light load. An ordinary centrifugal governor so adjusted would be so sensitive and unstable as to be wholly useless. By the use of a dashpot in connection with an inertia governor, however, it is possible to secure good stability with an adjustment of the spring that will cause the engine to run faster under load than it does when running light.

47. In practice, no steam-engine governors are built in which the relation between the distance of the center of gravity of the weight from the center of the shaft and the amount of deflection of the spring is as simple and direct as is shown in Fig. 21; in nearly all cases, however, the adjustment of the spring depends on the principles illustrated by that figure. In most governors the center of gravity of the weights and the point of attachment of the spring to the weights move in the arcs of circles having different radii, but the construction is generally such that the amount of deflection of the spring is very nearly directly proportional to the outward motion of the weights, and the relations between spring adjustment and sensitiveness of governor that were developed in Arts. **43–46** hold good. In some cases, however, the spring does not deflect proportionally to the outward movement of the weights; also, the effective length of the lever arms at which the weights or the springs act may vary so much during the movement of the weights as to materially change the relation between the action of the weights and springs and so affect the sensitiveness of the governor. For these cases no general rule for expressing the relation between spring tension and centrifugal force can be formulated. For such governors, each case must be considered by itself, and the particular kind of adjustment required to secure a given result must be determined by trial.

RULES FOR ADJUSTING THE SENSITIVENESS OF GOVERNORS.

48. A consideration of the principles developed in Arts. **43–46** shows that if the spring of a governor is so adjusted that its deflection is proportional to the distance of the center of gravity of the weights from the center of the shaft, the condition of isochronism will be fulfilled, and the governor will be too sensitive. To make the governor less sensitive, the spring must be so adjusted that its deflection for a given position of the weights is less, proportionally, than the distance of the center of gravity of the weights from the center of the shaft. In accordance with these principles we readily deduce the following rules for spring adjustment:

Rule 2.—*To increase the sensitiveness of the governor, increase the initial tension of the spring.*

Rule 3.—*To make the governor less sensitive, reduce the initial tension of the spring.*

In general, it is desirable to adjust the spring so as to make the governor as sensitive as is practicable without inducing hunting.

49. Adjusting for Changes of Speed.—From what has been said, it is evident that in order to change the speed of an engine with a shaft governor it is necessary to make such an adjustment as will change either the centrifugal force F of the weights or the resistance T of the springs. The resistance of the spring, and, in consequence, the speed of the engine, can readily be changed in most types of governors; a consideration of the principles developed in Arts. **43–46,** however, shows that a change in the spring tension has an effect on the sensitiveness of the governor. The best attainable sensitiveness will be secured with a certain initial tension, and any change from this will result either in making the governor sluggish or in making it too sensitive. For this reason changes in speed should never be made by changing the tension of the spring.

50. Since the initial spring resistance is not to be changed, it follows that a change in speed must be brought about by some change that will keep the force that balances the initial spring resistance, i. e., the centrifugal force F of the weights when in their inner position, constant. The formula for centrifugal force, $F = .00034\, W R n^2$, may be written in the form

$$n = \sqrt{\frac{F}{.00034\, W R}}.$$

A consideration of this formula shows that if F and R do not change, the only way in which the number of revolutions n can be varied is by changing the weight W. Since W appears in the denominator of the fraction that expresses the value of n, it follows that a given change in n requires a change of the opposite kind in W; in other words, if n is to be increased W must be made smaller, and vice versa. These principles may be expressed in the form of the following rules:

Rule 4.—*To increase the speed of an engine with a shaft governor, use lighter weights.*

Rule 5.—*To reduce the speed of an engine with a shaft governor, use heavier weights.*

51. Governors With Adjustable Weights and Spring Attachments.—Many governors have been made in which the weights and, in some cases, the points of attachment of the springs are adjustable along the arms by which they are connected to the pivots. This construction makes it possible to change the relative effects of the centrifugal force of the weights and the tension of the springs by changing the lengths of their lever arms with respect to the center of the pivot; the speed of the engine can thus be changed through a considerable range without changing the mass of the weights.

52. The type of adjustment mentioned in Art. **51** is illustrated by the **Buckeye governor,** Fig. 22. In the Buckeye engine a double valve is used. The main valve controls the events of admission, release, and compression and, since it is driven by a fixed eccentric, these do not vary with a change in the governor position. The cut-off is controlled by an auxiliary valve that is driven by the eccentric *C*. The point of cut-off is regulated by turning the eccentric around the shaft so as to vary the angle of advance without changing the throw. With a single valve, a change in the angle of advance without a change in the throw of the eccentric will make a corresponding change in the lead and the points of release and compression as well as in the point of cut-off.

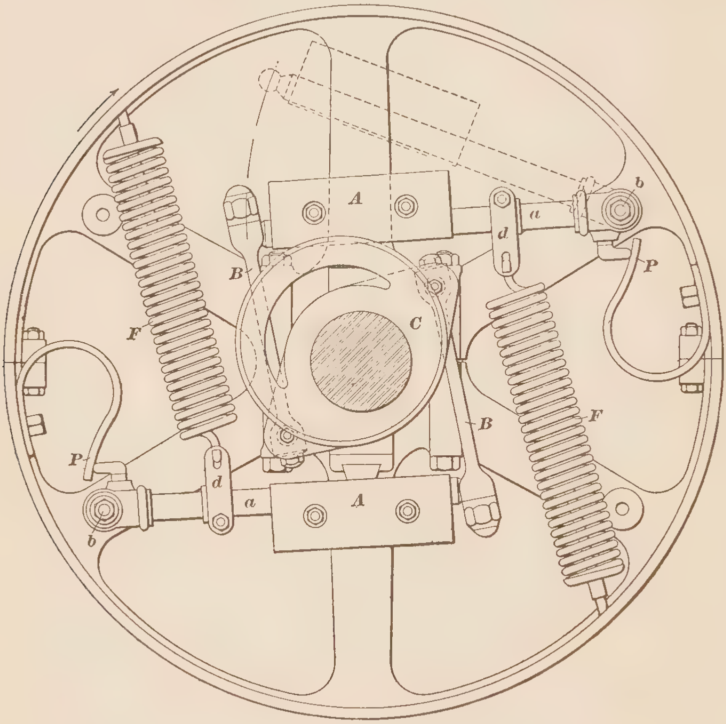

FIG. 22.

Also, with a single valve and an eccentric with a fixed throw, any change in the angle of advance that would be sufficient to give a satisfactory change in cut-off would vary the other events to too great an extent for practical use. With a double valve, however, in which the point of cut-off is the only event controlled by the valve driven by the adjustable eccentric, a change in the angle of advance without a change in eccentric throw gives very satisfactory results.

53. In the Buckeye governor, the weights *A* and the spring connections *d* of the main springs *F* may be shifted lengthwise along the arms *a*. These arms are pivoted to the governor case at the points *b* and are connected to the eccentric *C* by the links *B*. A study of the construction of this governor in conjunction with the principles of the lever shows that the effect of either the centrifugal force of the weights or of the tension of the springs will be increased by shifting their points of attachment farther from the pivot; we, therefore, readily deduce the following rules for changing the speed of an engine with a governor having adjustable weights or spring connections:

Rule 6.—*To increase the speed of the engine, shift the weight nearer the pivot, or shift the spring connection farther from the pivot.*

Rule 7.—*To reduce the speed of the engine, shift the weight farther from the pivot or shift the spring connection nearer the pivot.*

The amount of variation in speed that can be obtained by these adjustments will generally be small; if a greater variation is needed, it must be made by changing the mass of the weights in accordance with the rules 4 and 5.

54. In addition to the main springs *F*, the Buckeye governor, Fig. 22, is provided with two auxiliary springs *P*. These auxiliary springs act in opposition to the main springs until the weights have moved outwards through the first half of their range; during the outer half of the range of motion of the weights, the springs *P* cease to act and the weights are affected only by the tension of the main springs. The auxiliary springs acting in opposition to the main springs with a force that decreases from the beginning of the outward motion of the weights until mid-position is reached, make it possible to give the main springs a higher initial tension, and thus to secure a greater degree of sensitiveness during the outer half of the range of motion of the weights than could be obtained without causing unsteadiness and hunting during the inner half of the range.

GENERAL INSTRUCTIONS.

55. Considerable force is required to overcome the friction of the valves and valve gearing of most engines; the inertia of the valve and connections must also be overcome, that is, they must be started from rest and given their maximum speed during each stroke. The governor must be powerful enough to enable it to exert the force required to overcome these resistances, and to adjust itself promptly to small changes in speed; it must respond to changes in speed with the greatest of freedom, but it must not be seriously affected by the great variations in the force required to move the valve at different parts of the stroke. The fulfilment of these conditions makes necessary the use of heavy weights and powerful springs, with the result that heavy pressures are developed at the pivots. In order that the friction due to these pressures may not become serious, it is necessary to provide special means for securing good lubrication of the pivots, or to use bearings that will work freely without lubrication. In the later types of governors, particularly those of the inertia type, the number of joints has been reduced to a minimum and roller bearings have been substituted for plain bearings. The roller bearings require no lubrication and the governors will work with but little attention. Knife-edge bearings are also used with excellent results in those positions where the pressures are always in the same direction. If the joints are not properly constructed and lubricated, the excessive friction will soon greatly reduce the promptness with which the governor will respond to changes in speed and it will tend to hunt.

56. In order to prevent the varying force required to drive the valve at different parts of its stroke from disturbing the governor weights, a number of so-called self-locking governors have been built. In these governors, the motion of the weights is transmitted to the eccentric through a self-locking mechanism; that is, a mechanism that, like the screw or wedge, will enable the force exerted by the weights to move the eccentric, but will not permit a force acting on the eccentric to affect the weights. A disadvantage of governors of this type is that the frictional resistances, especially when lubrication is not attended to with the greatest care, are excessive; they are, therefore, likely to be sluggish and unsteady. The disturbing effects of friction often produce more serious results than would have been developed by the forces that the self-locking mechanism is designed to overcome.

VALVE GEARS.

(PART 1.)

DEFINITION AND CLASSIFICATION.

DEFINITION.

1. The **valve gear** of an engine may be defined as that part of the mechanism which acts to control the distribution of steam to the cylinder. In the plain slide-valve engine, the valve gear consists of the eccentric, eccentric rod, rocker-arm or valve-stem guide, valve stem, and the valve. In the different types of reversible engines, or variable and automatic cut-off engines, the valve gear includes the mechanism by means of which the motion of the valves is governed so as to change the direction of motion of the engine or vary the amount of work done in the cylinder.

GENERAL CLASSIFICATION.

2. Valve gears are classified in a variety of ways, depending partly on their type and partly on the kind of service for which they are used. The most common terms used to denote the different types and classes of service are the following:

3. Fixed Cut-Off Gears.—In this type the motion of the valve does not vary with the amount of work to be done in the cylinder. With a given setting of the valve, cut-off

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is constant. A plain slide-valve engine is the most familiar example of an engine fitted with a fixed cut-off gear. With it the points of admission, cut-off, release, and compression do not vary.

4. **Variable Cut-Off Gears.**—Under this head are included all those gears in which the amount of work done in the cylinder can be regulated while the engine is in operation by varying the motion of the valve so as to change the point of cut-off. If the motion of the valve is controlled automatically by some form of governor, a variable cut-off gear becomes an **automatic cut-off gear.**

5. **Positive-motion gears** are those in which the valve is directly controlled by the eccentric and governing mechanism throughout its whole range of motion.

In **cam gears** the valves receive their motion from cams instead of eccentrics. Cam gears are much used in connection with poppet valves.

6. **Releasing gears,** or **trip gears,** as they are sometimes called, vary the point of cut-off by releasing the admission valve from the control of the eccentric, and thus permit it to be quickly closed by some other device, usually a spring or dashpot. The Corliss valve gear is the most familiar type of releasing gear.

7. **Reversing gears** are used to change the direction of motion of the engine; in most cases they may also be used to vary the point of cut-off; hence, most reversing gears are also variable cut-off gears. There are two principal types of reversing gears, viz., link motions and radial gears.

8. The classification may be further extended to make it cover the number and type of valves used to control the steam distribution. In accordance with the number of valves, the leading divisions are **single valve, double valve,** and **four valve.** The leading types of valves are **slide valves, rotary valves,** and **poppet valves.**

CUT-OFF, OR EXPANSION, VALVES.

9. Purpose of Cut-Off Valves.—In order to extend the range of cut-off beyond that which may be obtained with the plain slide valve driven by a fixed eccentric and at the same time not affect the events of admission, release, and compression, various types of auxiliary valves, generally known as **expansion valves,** or **cut-off valves,** have been designed for use in conjunction with the plain slide valve. These auxiliary valves are generally driven by a separate eccentric; their sole purpose is to stop the flow of steam to the cylinder at the desired point in the stroke, leaving the other features of steam distribution to be controlled by the main valve in the usual manner. Since the point of cut-off is the only event controlled by the auxiliary valve, any change in the proportions of this valve or in the position of its eccentric that will secure the desired change in cut-off may be made without affecting the other events in steam distribution; consequently, the point at which the auxiliary valve cuts off the supply of steam may be varied in any convenient way; for example, the lap of the valve or the angle of advance or eccentricity of the auxiliary eccentric may be changed as may be most convenient.

10. A simple application of the cut-off valve is shown in Fig. 1. The main valve V is a plain slide valve driven by a fixed eccentric in exactly the same manner as in the plain slide-valve engine. This valve is given such proportions as will secure the desired points of admission, release, and compression, care being taken that the lap is such that it will not cut off steam earlier in the stroke than the latest point at which it may be desired to have cut-off occur, usually about $\frac{3}{4}$ stroke.

The main valve is located in a section S of the steam chest that is separated from the remainder by a plate or partition p. This partition has ports or openings t, t that form passages connecting the two chambers, and it also forms the valve seat of the auxiliary valve v. The auxiliary valve has ports *s*, *s*. It is driven from the main shaft by an auxiliary eccentric. When it is in mid-position, as shown in the figure, the ports *t*, *t* are open and allow steam to fill the chamber around the main valve. As it moves either way from the mid-position, it covers the ports *t*, *t*, and so shuts off the supply of steam from the main valve.

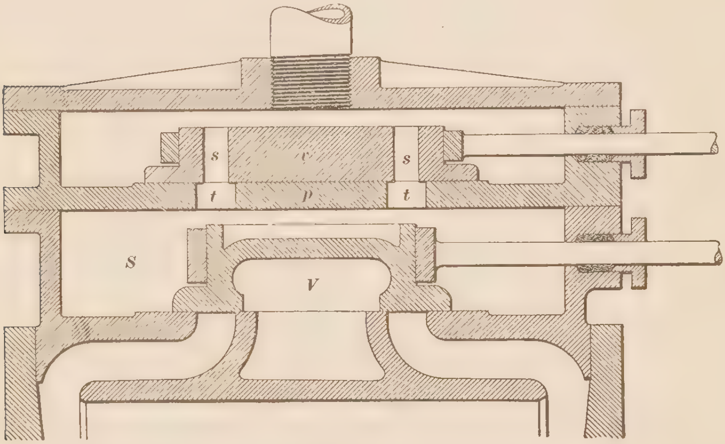


FIG. 1.

11. The main valve is set in exactly the same manner as any plain slide valve. The auxiliary valve must be so proportioned and set that it will open the ports *t*, *t* as soon as or before the main valve opens the main port, but they must not be opened before the cut-off occurs with the main valve, otherwise steam will be readmitted to the cylinder and the effect of cut-off will be destroyed. The point of cut-off may be varied by changing either the eccentricity or angle of advance of the auxiliary eccentric, the latter being the more common method. If, as has sometimes been done, but one port *t* is used, and the valve is made of two plates, the cut-off may be varied by changing the distance between the edges of the plates forming the auxiliary valve. By either of the first two methods, the practical range of cut-off is small; by the third, a greater range can be obtained. A disadvantage of this type of expansion valve is that when cut-off takes place, the space in the main-valve chamber is filled with steam. This has the effect of increasing the clearance space during the period of expansion, and thus destroys some of the benefits derived from the earlier cut-off. On account of these limitations and disadvantages, this simple form of expansion valve which, from the name of its designer, is known as the **Gonzenbach valve,** is but little used to-day.

12. The **Meyer cut-off valve,** Fig. 2, is an improvement on the preceding type and is free from most of its disadvantages. The main valve consists of a flat plate with a chamber *E* on the under side. It acts in exactly the same manner as a plain slide valve, from which it only differs in that the ends of the valve are extended so as to form passages or ports *A*, *A* leading to the edges that control the admission of steam to the cylinder. The cut-off valve consists of two plates *B*, *B* that slide on the top of the main valve so as to close the passages *A*, *A* at the point at which it is desired to have cut-off take place. The main valve is driven from its eccentric by the valve stem *D* and the cut-off valve from a separate eccentric by the valve stem *C*. The main eccentric is set in the same manner as with the plain slide valve. The auxiliary eccentric is so set that, when cut-off takes place the cut-off valve has a motion opposite in direction to that

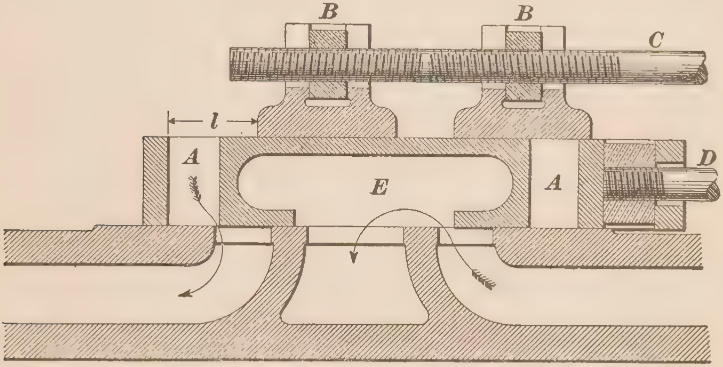


FIG. 2.

of the main valve. The cut-off valve thus closes the passage *A* through which steam is being admitted to the cylinder. The action may be studied by considering Fig. 3 in connection with Fig. 2.

13. In Fig. 3, *O a* represents the crank in the dead-center position, corresponding to the valve position shown in Fig. 2, while *O b* and *O c* represent the corresponding positions of the main and auxiliary eccentrics. It will be noticed that the main eccentric *O b* is set in its usual position of 90° + angle of advance ahead of the crank, thus giving the main valve the amount of lead shown in Fig. 2. The auxiliary eccentric *O c* is set in a position nearly opposite to that of the crank. By inspecting the two figures, we see that as the crank moves in the direction shown by the arrows, the main valve, driven by the eccentric *O b*, will move to the right and open the left-hand port; at the same time the cut-off valve, driven by the eccentric *O c*, will move to the left and it will close the passage *A* as soon as the combined motion of the two valves is equal to the distance *l*, Fig. 2.

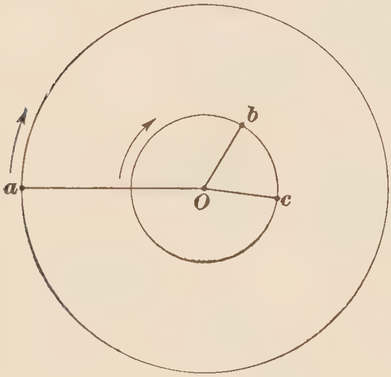


FIG. 3.

The valve stem *C* is provided with right- and left-hand threads that work in threaded plates or nuts in the valve plates *B*, *B*. By turning *C* in one direction or the other, the distance between the outside edges of the plates *B*, *B* can be increased or diminished. If the distance between the plates is increased, the distance *l* becomes shorter and cut-off takes place earlier; the opposite motion makes cut-off take place later. As usually arranged, the rod *C* may be turned by a hand wheel placed outside of the steam chest, so that cut-off may be changed while the engine is running.

14. A form of riding cut-off valve that is somewhat simpler than the Meyer valve is sometimes used. In this form the cut-off valve consists of a single plate, the outside edges of which close the passages through the main valve in the same manner as the edges of the two plates of the Meyer valve. Since the distance between the outer edges of this cut-off valve is constant, the point of cut-off can be changed only by a change in the eccentric, the usual method being to vary the angle of advance.

15. Setting the Riding Cut-Off Valve.—The first step in setting a riding cut-off valve is to examine the main valve to see that it is properly set; that is, try the lead at both ends and set the valve if it is unequal. The cut-off valve should then be so adjusted that it will travel equal distances on each side of its central position, when referred to its seat on the back of the main valve. If the cut-off valve is of the non-adjustable type described in Art. **14,** set its eccentric in the following manner: Turn the engine to the point at which it is desired to have cut-off take place; then turn the cut-off eccentric until the edge of the cut-off valve just closes the passage of the main valve that is open to the cylinder; the eccentric may then be fastened to the shaft.

16. To set the Meyer valve, the first step is to adjust the length of the eccentric rods in the same manner as for the non-adjustable type. The screw of the valve stem should then be turned so as to separate the plates to the greatest distance at which they are designed to be worked; stops are often provided to limit this distance between the plates. To set the eccentric, turn the engine to the point at which it is desired to have earliest cut-off take place, turn the eccentric on the shaft until the cut-off valve just closes the passages through the main valve, then fasten the eccentric to the shaft.

17. The Meyer valve is sometimes used to secure a variable cut-off with a reversing engine; when so used, if the main eccentrics are given equal angles of advance and the cut-off eccentric is set exactly opposite to the crank, the cut-off will be the same for either direction of motion of the engine. It is, however, sometimes important to secure a certain range of cut-off for one direction of motion that cannot be secured by such a setting of the eccentric. For such a case, the valves are set with special regard to the more important direction of rotation.

A modification of the Meyer valve is shown in Fig. 4. Here the cut-off valve consists of a single plate *a* with inclined edges working over a main valve *b* having passages similarly inclined. The cut-off valve stem, in addition to the endwise motion given to it by its eccentric, may be rotated. It carries a pinion *p* that works in a rack on the cut-off valve. By this means the cut-off valve may be moved across its seat on the main valve; this will have the same effect on the effective distance between the edges of the cut-off valve and the edges of the ports through the main valve as the changes made by the right- and left-hand screws in the distance between the plates of the Meyer valve.

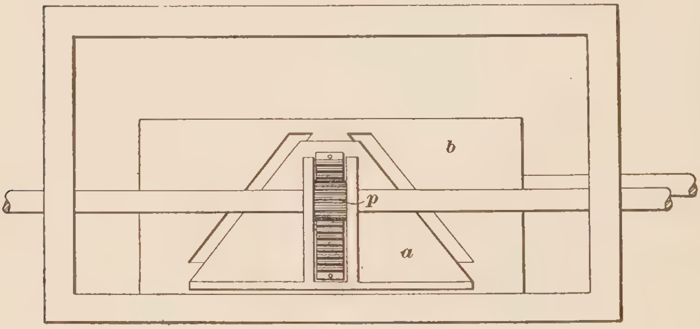


FIG. 4.

Another somewhat similar modification is made by giving the cut-off valve and its seat a circular form; in this case the ends of the cut-off valve and the openings through the main valve are helical, their shape being the same as though the valve and seat of Fig. 4 were wrapped around a cylinder. By rotating the cut-off valve around its center by means of the valve stem, the same effect is produced as by sliding the cut-off valve in Fig. 4 across its seat. The turning motion of the valve stem in the two types of cut-off valves here described is often accomplished by a governor, thus producing an engine with an automatic cut-off.

VALVES FOR SINGLE-VALVE AUTOMATIC CUT-OFF ENGINES.

INTRODUCTION.

18. It is a well-known fact that the action of a shaft governor is materially affected by the force required to drive the valve; a plain **D** slide valve, requiring as it does a great driving force, not only absorbs considerable of the power of the engine, but it has a disturbing effect on the action of the governor that, especially with large engines, becomes so serious as to make the use of this type of valve with shaft governors impracticable. To relieve the governor as much as possible from the effects of the frictional resistances incident to the **D** slide valve, a great number of balanced valves, of which the piston valve is one type, have been invented. When used in conjunction with a shaft governor that varies the throw of the eccentric, the plain slide valve has another defect that is not overcome by the mere process of balancing. This defect is that the reduced travel of the valve consequent on the short throw of the eccentric at early cut-offs has the effect of greatly restricting the port opening. At high speeds such a reduction in port opening causes an imperfect filling of the cylinder with live steam; that is, the steam is wiredrawn in its passage through the partly opened ports. This difficulty can be largely overcome by increasing the travel of the valve so as to give it a considerable amount of **overtravel** for late cut-offs. By overtravel is meant that the valve is given a travel greater than is required to give a full port opening of the steam ports. Such a remedy, however, involves other undesirable conditions.

To provide for the overtravel, the central chamber of the valve and the bridges between the steam and exhaust ports must be made wider; this means a larger and heavier valve. The attempt to secure a liberal port opening with early cut-offs by giving the valve overtravel thus involves an increase in the travel of the valve and in its weight, both of which add to the power required to move it. Once during each stroke the heavy valve must also be started from rest and given a velocity that becomes greater as the distance through which the valve must move is increased; to overcome the inertia of the valve, a very considerable force is required, and this force, which must act through the medium of the governor, and in consequence tends to disturb the governor action, is increased by the attempt to secure a liberal port opening by means of an increase in the valve travel.

Several designs for shaft-governor engines secure an increased port opening by the use of multiple-ported valves, in which steam enters the cylinder through two or more passages. While the travel of the valve may be reduced by this means, its size and weight are generally increased, and this neutralizes the advantage of short travel to a considerable degree.

THE PISTON VALVE.

19. The **piston valve** is one of the lightest and most perfectly balanced types that has yet been devised, and it is readily made double-ported. It has, however, several features that have restricted its use to a very great extent. Unless it is provided with some form of packing there is no means of adjustment to provide for wear; consequently, it soon becomes leaky and wastes steam badly. In vertical engines, where the weight of the valve is not supported by the seat, the wear is not so serious, and there are many cases of such engines in which piston valves with no packing except grooves in which water collects have given the best of service. In a horizontal engine, however, the weight of the valve invariably wears the lower part of the seat and destroys its circular form. Packing rings of some form must here be provided or the valve will soon leak. In some cases the valve seat consists of a bushing that can easily be removed when worn; in order that a good fit between valve and bushing may be secured, it will generally be necessary to renew the valve as well as the bushing, or to fit the old valve and new bushing to each other. The long ports required to surround the piston valve make the clearance large. This type of valve as generally constructed offers no relief to water that may be caught in the compression space after the exhaust port closes.

THE FLAT PRESSURE PLATE VALVE.

20. The **flat-pressure plate valve** is a type that has many of the advantages of the piston valve and at the same time overcomes some of its faults. Fig. 5 shows a form of this valve used in the engines built by the Ames Iron Works, and is a form that, with several minor modifications, is used in a great number of shaft-governor engines. The valve *V* consists of a thin rectangular casting with openings or ports through it for the passage of live and exhaust steam to and from the cylinder. It works between the face of the valve seat and a similar face formed by a heavy pressure plate *P*. The pressure plate is separated from the valve seat by strips *s*, *s'* that are made just enough thicker than the valve to permit the valve to slide freely between the pressure plate and the valve seat. When the engine is working, the pressure plate is held in place against the strips *s*, *s'* by the steam pressure; when steam is shut off, the spring *N* prevents the plate from falling away from its position against the strips. This construction permits the valve and pressure plate to act as a relief valve for the escape of water from the cylinder; an excessive pressure in the cylinder will lift the valve and plate and compress the spring. The danger of breakage from water in the cylinder is thus somewhat reduced; it is not, however, a perfect protection; with quick rotative speeds, the pressure often

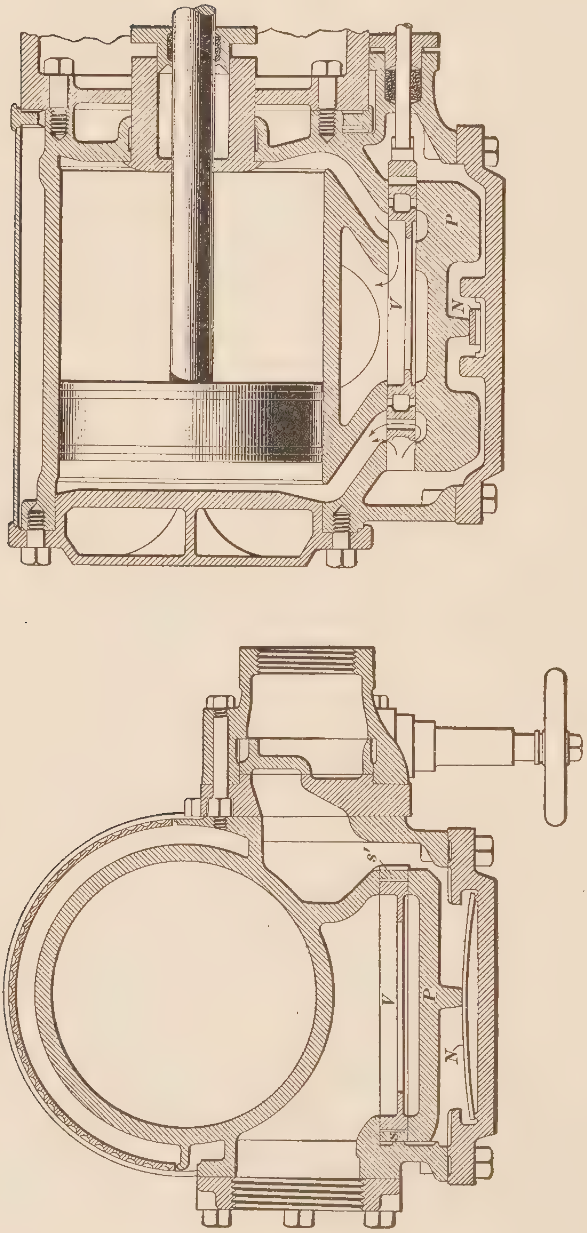


FIG. 5.

becomes great enough to seriously damage the engine before water can be forced out through the ports and overcome the inertia of the valve and pressure plate.

21. An advantage of the pressure-plate valve over the piston valve is that in horizontal engines the wear due to the weight of the valve, coming as it does on the lower edge and the strip *s'*, does not affect the tightness of the valve. Another advantage is that when the valve wears so as to allow steam to leak between it and the faces of the valve seat and pressure plate, the strips *s*, *s'* can be planed or scraped so as to overcome this wear; this operation, however, requires considerable care and skill. If the valve or the faces between which it works are unevenly worn, it will be necessary to scrape them to a new surface; this is a tedious and somewhat difficult task with the facilities available in most engine rooms.

With either piston valves or pressure-plate valves, the precaution is required when starting the engine to admit steam to the steam chest slowly so as to warm the valve and its seat gradually and evenly; if steam is turned on too rapidly, there is danger that the valve will be heated faster than its seat and swell so much as to bind or seize in its seat and break some part of the valve gearing. To reduce this danger, one make of engine using a pressure-plate valve, the **Woodbury,** is provided with a device for lifting the pressure plate a short distance from the valve until the valve and its seat become uniformly heated.

SELF-ADJUSTING BALANCED VALVES.

22. Methods of Balancing.—Many of the faults of the valves described in Arts. **19** to **21** are overcome by the use of self-adjusting balancing devices, of which there are several types. In one the pressure plate is held against the back of the valve and the valve against its seat by a certain amount of unbalanced steam pressure. In another form that has been much used on locomotives and some types of stationary engines, although not to so great an extent on shaft-governor engines, the back of the valve is fitted with packing rings or strips that bear against a fixed pressure plate and prevent live steam from reaching the space between the pressure plate and the back of the valve. Fig. 6 shows an

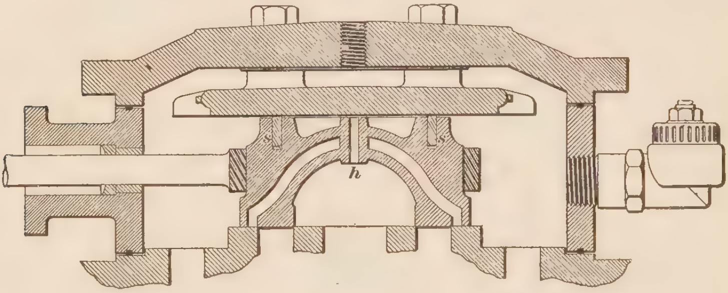


FIG. 6.

Allen valve with this method of balancing. The packing strips *s*, *s* are fitted in grooves in the back of the valve and held out against the pressure plate by light, flat springs. A hole *h* connects the space at the back of the valve with the exhaust cavity; the pressure in this space is, therefore, the same as the pressure in the exhaust port.

23. The **Ball valve**, Fig. 7, is a form of self-adjusting balanced valve that is made of two separate pieces, one working telescopically in the other, the joint between the two being kept steam-tight by packing rings. Each part of the valve bears against its own seat and controls the flow of steam through its own set of ports. Live steam enters the chamber *a*, formed by the combination of the two parts, through the opening in the upper wall of the steam chest and passes through the ports to the cylinder; the exhaust is discharged into the steam chest around the outside edges. The valve is thus seen to be *indirect* and the combination of the two parts, each with its separate ports, secures a double-ported effect for both steam and exhaust. The two parts of the valve are held against their respective seats by the pressure of the steam in the chamber *a*, the action of this pressure tending to separate the parts. By this arrangement the valve automatically follows up any wear, and if the working faces are kept in good order it will always be tight. It is not quite as perfectly balanced as the piston valve or the flat-pressure plate valve of the type described

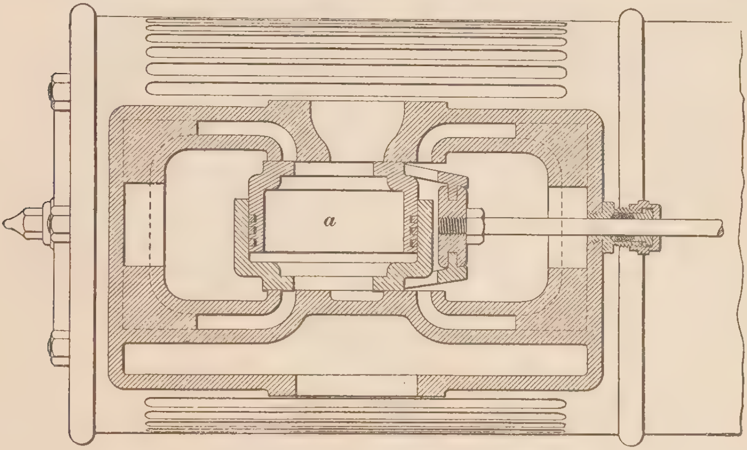


FIG. 7.

in Arts. **20** and **21**, and its weight is probably a little greater than either of these types. With it the ports are necessarily long and the clearance space is large; in this respect it is probably to be compared with the piston valve.

THE CORLISS VALVE GEAR.

CHARACTERISTIC FEATURES.

24. The most characteristic features of the Corliss valve gear are four in number :

1. *The Four Oscillating Valves.*—Two of these valves are placed at each end of the cylinder—one (the admission valve) controlling the admission of the steam, the other (the exhaust valve) controlling the liberation of the expanded steam.

2. *The Wristplate.*—This is an oscillating plate, disk, or lever actuated by the eccentric on the main shaft and in turn communicating motion to the four valves.

3. The *releasing mechanism*, of which the governor forms a part. This disconnects an admission valve from the wristplate when the supply of steam is to be discontinued through its port.

4. *The Closing Apparatus.*—This, immediately upon the release of an admission valve, brings the valve back to its closing position.

Corliss valve gears have been built in a great variety of forms, differing from one another principally in the details of the releasing mechanism and in minor features of construction. These gears, however, all operate on the same general principles, and when one is understood there will be little difficulty in understanding any of the others.

CONSTRUCTION.

25. Cylinder and Valves.—Fig. 8 is a section that shows the general construction of one form of cylinder and valves. As shown, there are four distinct valves. Two of these, v' and v, admit steam to the crank end and head end of the cylinder, respectively. They are in communication with the steam chest d and steam pipe s. The other two valves, r and r', communicate with the exhaust chamber l and exhaust pipe o. The valve r' allows the steam to exhaust from the crank end of the cylinder; r controls the exhaust from the head end.

The valves v, v' are called the **steam valves,** and r, r' are called the **exhaust valves.**

The valve faces are cylindrical in form, and the valves rotate on circular valve seats. Each valve extends across the cylinder; that is, the length of the valve is about equal to the cylinder diameter.

An inspection of Fig. 8 shows that the steam valves open their ports when their faces move away from the center of the cylinder towards the heads. It is also seen that both the steam and the exhaust valves are *single-ported;* that is, they open but one passage for the flow of the steam to and from the cylinder.

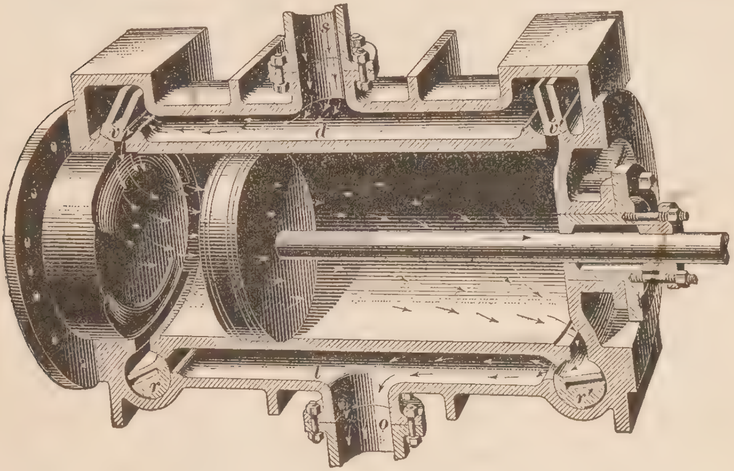


FIG. 8.

26. Double-Ported Valves.—Fig. 11 shows a method of construction by means of which the valves are all made double-ported. The steam valves in this case open their ports by rotating in the opposite direction to that in which the steam valves of Fig. 8 rotate; that is, to open the ports the valve faces move away from the heads towards the center of the cylinder. The form of the valves is such that steam flows past two edges, as shown by the small arrows around the steam valve *a* and the exhaust valve *e'*. This construction provides a sufficient port opening with a much smaller valve movement and greatly improves the distribution of steam to the cylinder.

MECHANISM FOR OPERATING THE VALVES.

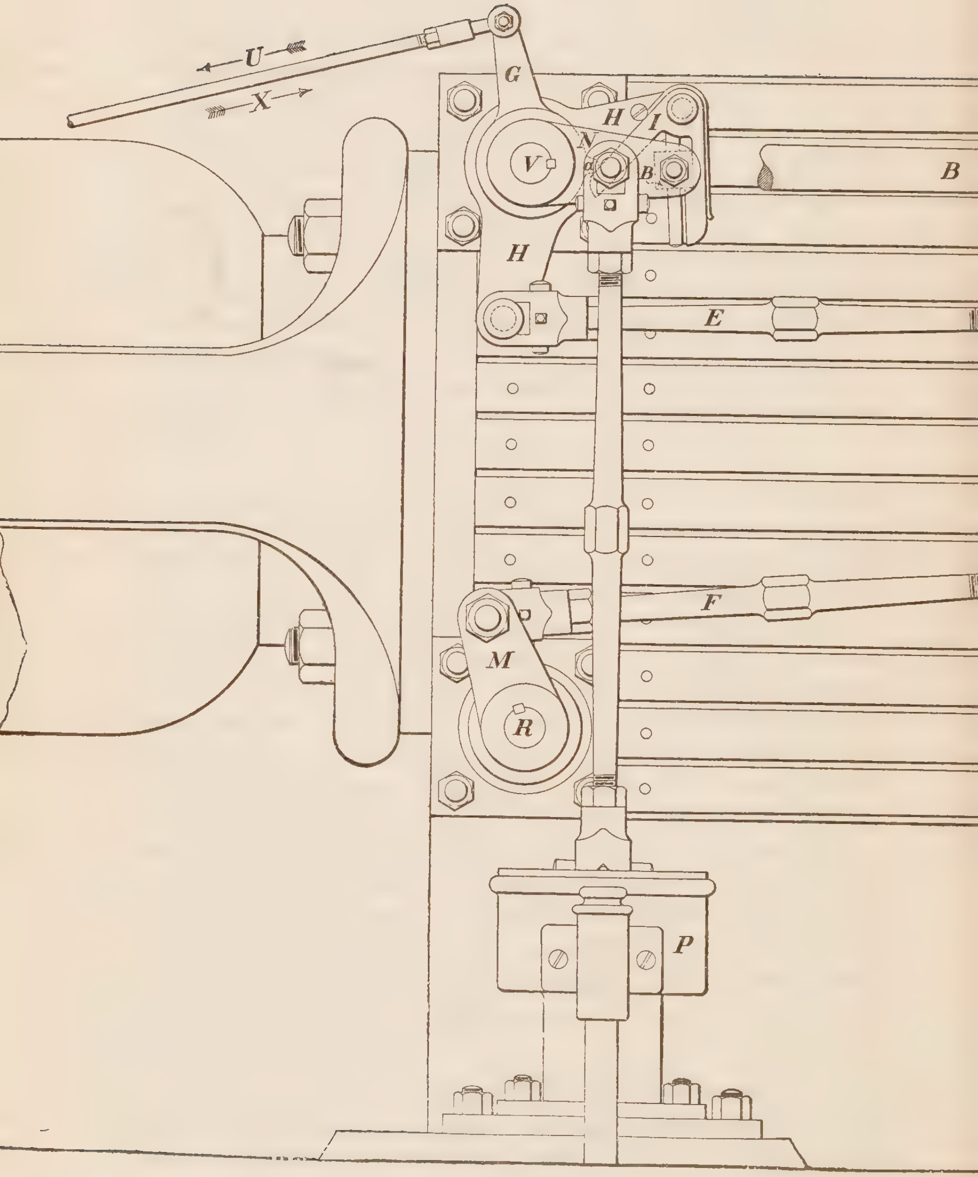
27. Fig. 9 shows the general arrangement of one of the most familiar forms of the mechanism for operating the valves. The steam valves are driven by the spindles *V* and *V'*, called the **steam-valve stems**, and the exhaust valves by the spindles *R* and *R'*, called the **exhaust-valve stems**; the center lines of these spindles coincide with the center lines of their respective valves.

To the steam-valve stems *V*, *V'* are keyed cranks, as *N*. In the figure, the crank keyed to *V'* is removed to show more clearly the disengaging link and hook *I'*. Likewise, to the exhaust-valve stems *R*, *R'* are rigidly keyed the cranks *M*, *M'*.

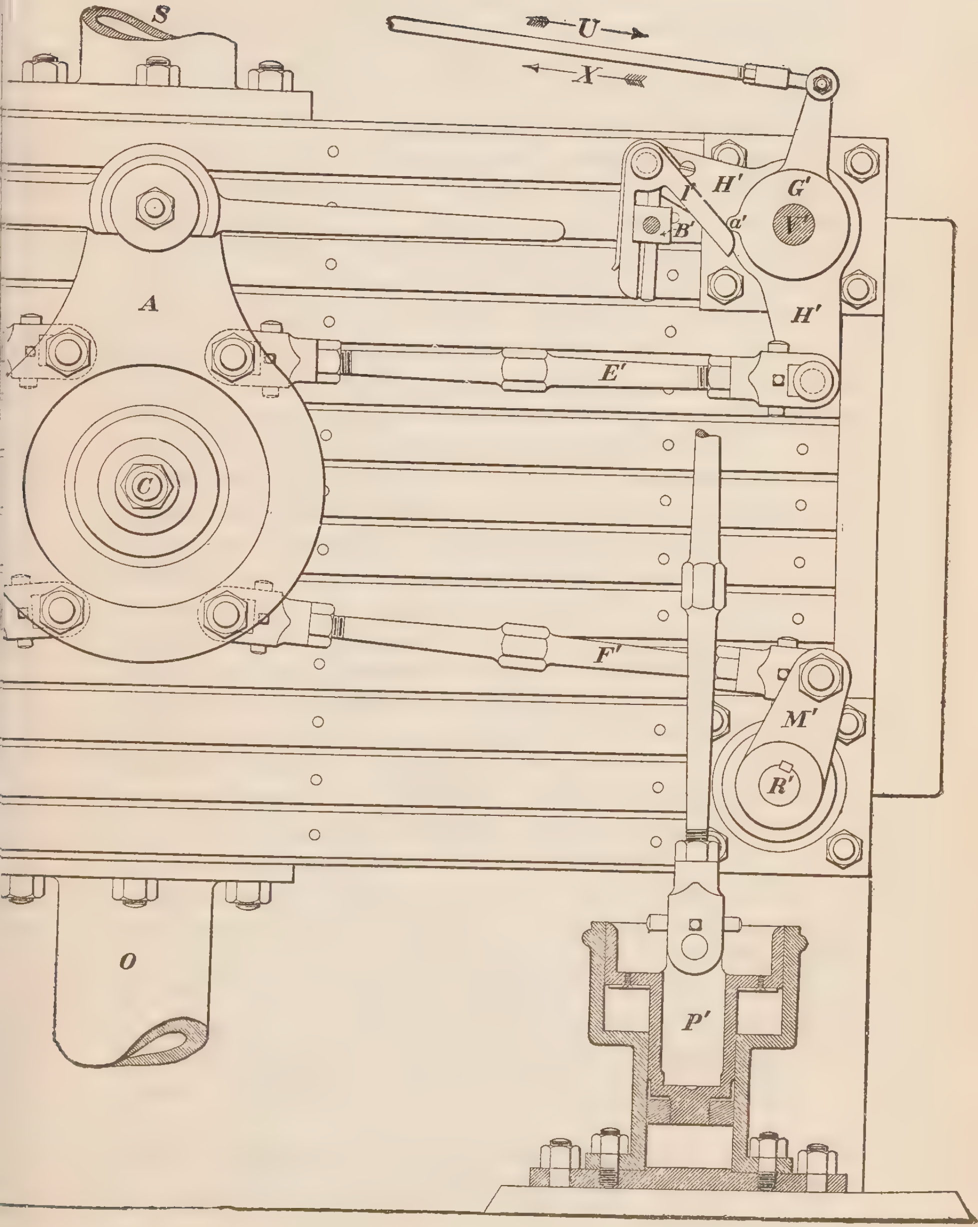
The disk, or **wristplate**, *A* is made to rock on the stud *C* by the wristplate rod *B*, which takes its motion from a rocker-arm that in turn is driven through an eccentric rod by an eccentric placed on the main shaft. To the plate *A* are connected the four **valve rods** *E*, *E'*, *F*, and *F'*. As the plate rocks, these valve rods are given a to-and-fro motion.

28. The exhaust-valve rods *F* and *F'* are connected directly to the cranks *M* and *M'* of the exhaust valves. The motion of the plate *A* is thus communicated to the exhaust valves, causing them to open and shut the exhaust ports at the proper time. The steam-valve rods *E*, *E'*, however, are not connected directly with the cranks keyed to *V* and *V'*, but with the bell-cranks *H*, *H'*, called the **admission cranks.** These bell-cranks can rotate around the valve stems *V*, *V'*, not being connected rigidly to them. The bell-cranks carry the disengaging links *I*, *I'*, which, therefore, have the rocking motion given to the bell-cranks by the rods *E*, *E'*. These disengaging links hook on to blocks *B*, *B'*, rigidly fastened to the cranks keyed to *V* and *V'*. Consequently, when the bell-cranks *H*, *H'*, and with them the links *I*, *I'*, are rotated by the motion of the valve rods, the cranks, as *N*, are forced to move with them. This motion of the crank rotates the valve and admits steam to the cylinder.

29. Cut-Off Mechanism.—On the valve stems *V* and *V'* are collars *G* and *G'*, called the **trip collars**; these collars are free to rotate on the stems. Each collar is provided with



FIG


S
U
X
H′
G′
I′
a′
B′
V′
A
H′
E′
C
F′
M′
R′
O
P′

an arm to which is attached a rod UX, called a **governor rod,** that leads to the governor. Each collar also has a projection, as a', called a **trip,** that lies in the same plane as the arms I and I' of the disengaging hooks. Considering now the right-hand admission valve, imagine the wristplate A to be rotating towards the left, in a direction opposite to the direction of motion of the hands of a watch; this motion will be transmitted by the rod E' to the crank H' and will carry the link and hook I' in an upward direction, thus lifting the block B' and rotating the right-handed crank to which B' is attached, which, it will be remembered, was removed to show the other parts more clearly. This crank, being keyed to the stem V', turns this stem and so opens the valve, the direction of motion of the valve face being the same as with the valves described in Art. **26.** As the link I' is lifted, its arm strikes the trip a' on the collar G'; this swings the link I' about the pin by means of which it is pivoted to the crank H' and thus detaches the hook from the block B' and stops the opening motion of the valve. To the cranks keyed to V and V' are attached rods leading to the pistons of the dashpots P and P'. When the crank is rotated by the upward motion of the block B', as just described, the dashpot rod attached to it lifts the piston in P' and creates a partial vacuum in the space below the piston. When the block B' is released from the hook on I', the pressure of the atmosphere forces the dashpot piston downwards, thus quickly closing the valve and cutting off the supply of steam to that end of the cylinder.

When the wristplate A moves in the opposite direction to that just considered, the hook I' is moved downwards until it again engages the block B' and the motion above described is ready to be repeated upon the reversal of the wristplate. The action of the valve for the crank end of the cylinder is precisely the same as for the head end, except that it is opened when the wristplate rotates towards the right.

30. Analysis of the Wristplate Motion.—As will be seen by a study of Fig. 10, the wristplate modifies the effect of the eccentric motion in such a way that the motion of the valves is considerably improved. In the figure, *O* represents

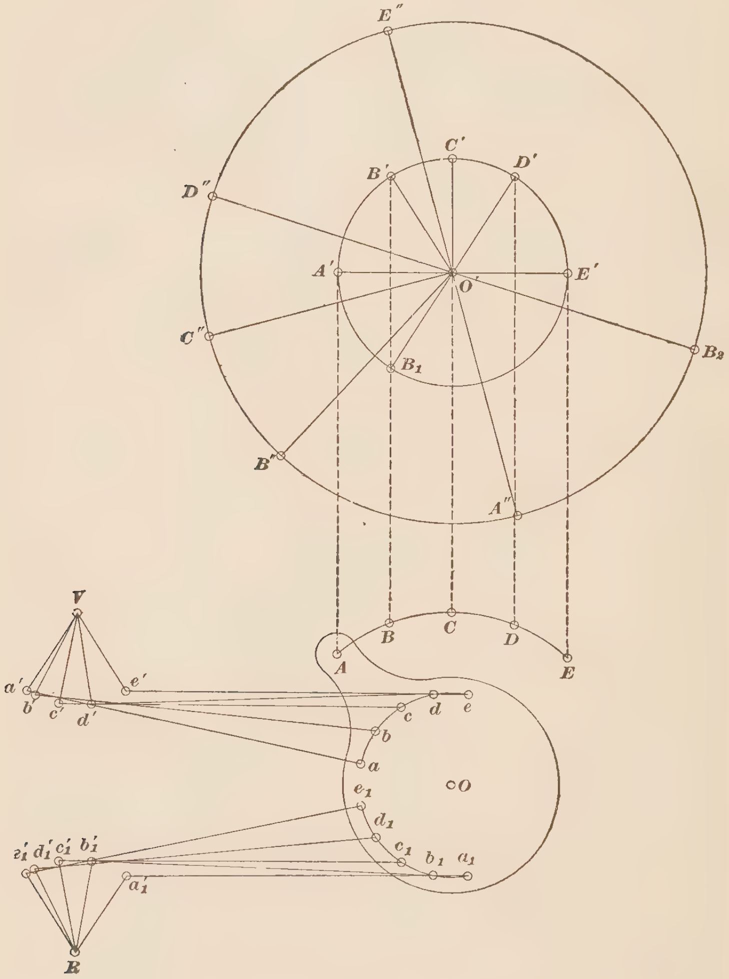


FIG. 10.

the center of the wristplate stud and *V* and *R* represent, respectively, the centers of the steam- and exhaust-valve stems for one end of a Corliss engine gear similar to that shown in Fig. 9. A, B, C, D, and E represent five positions of the pin on the wristplate to which the carrier rod is attached; of these positions, A and E are the extreme positions of the pin and the others are so chosen that the angular distances between two consecutive spaces are all equal. The positions of the steam-valve rods corresponding to the positions A, B, C, D, and E are represented, respectively, by the lines $a\,a'$, $b\,b'$, $c\,c'$, $d\,d'$, and $e\,e'$, and the similar positions of the exhaust-valve rods by $a_1\,a_1'$, $b_1\,b_1'$, $c_1\,c_1'$, $d_1\,d_1'$, and $e_1\,e_1'$. It is clearly shown by the figure that the motion imparted to the valves for a given angular motion of the wristplate varies through wide limits. For example, when the wristplate pin moves from A to B, the steam valve moves through the very short distance represented by $a'\,b'$; when, however, the wristplate pin reaches the middle of its motion, the motion imparted to the valve is much more rapid, as is seen by comparing the distances $C\,D$ and $c'\,d'$ with $A\,B$ and $a'\,b'$.

31. In order to compare the valve motion with the motion of the eccentric and crank, the eccentric and crank positions corresponding to the wristplate positions from A to E are drawn with a center O' vertically over the center O of the wristplate stud. The eccentric center is represented in the six positions, B_1 to E', of which A' and E' are the extreme positions of its throw and the corresponding crank positions are represented by the points on the crankpin circle lettered from B_2 to E''. Beginning now with the crank position B_2 and following the crank around in the direction of the arrow, it is seen that while the crank moves from B_2 to B'', a motion that corresponds to a motion of the piston through nearly $\frac{3}{4}$ of its stroke, the end of the valve crank has moved only from b' to its extreme position a' and back to b' again; during this large portion of the piston stroke the valve has been almost at rest. As the crank passes the position B'' and passes its dead center, the motion of the valve is seen to increase rapidly. This has the effect of opening the valve rapidly just as the piston reaches the end of its stroke, thus permitting a free admission of steam. The valve continues to open at a good rate of speed until it is closed by the releasing gear or until the eccentric reaches its extreme throw position E'.

32. By comparing the motion of the steam and exhaust valves, it is seen that the period during which the steam valve is nearly at rest corresponds to the period when the motion of the exhaust valve is most rapid. The exhaust valve was opened wide before the piston began its stroke to the left, and the period of most rapid closure is that during which the crank passes B'', corresponding to the point in the stroke at which compression begins. Following the point of exhaust closure, the motion of the valve is very small until the piston approaches the opposite end of its stroke, where release is to take place.

33. Varying the Point of Cut-Off.—An inspection of Fig. 9 shows that the point at which the block B' is released from the hook on I', and, in consequence, the point at which cut-off takes place, depends on the position of the trip a'. If the collar G' is rotated towards the left so as to move a' downwards, the arm of I' in its upward motion will strike a' and thus release the block B' earlier; the opposite motion of the collar will evidently have the contrary effect. We thus see that the point of cut-off can be varied by swinging the collars G and G'. The position of the collars is generally controlled by a pendulum governor acting through the rods $U X$. When the rods are moved in the direction of the arrows X, the trips are lowered and cut-off takes place earlier; motion in the direction of the arrows U raises the trips and causes a later cut-off.

34. Limitations to Range of Cut-Off.—When the engine is working under a light load, the governor collar is kept in such position by the governor that the cut-off trip releases the hook early in its motion, while for a heavier load the reverse is the case. As, however, the motion of the hook is an oscillating one, it can evidently not be released by the trip after it has started its return stroke, and, therefore, if the load on the engine becomes so heavy that the governor moves the trip so far that it will not be struck by the hook before the reverse motion of the hook begins, release of the hook will not take place at all. In this case the admission valve will be closed as if positively connected to the admission crank, the same as the exhaust valves.

35. Action of the Dashpot.—As shown in the sectional view, Fig. 9, the dashpot consists of two concentric cylinders of different diameters. In the upper and larger of these cylinders fits a piston having a central projecting portion that extends downwards and forms a plunger that fits the lower and smaller cylinder. The upper cylinder has an opening in its end, not shown in the figure, that may be regulated by a valve or cock. This opening is placed a short distance above the lower edge of the piston when it is in its lowest position; it allows a small quantity of air to enter as the piston rises, and this air checks the downward motion to an extent depending on the amount of opening of the valve. After the lower edge of the piston descends until the opening is covered, the further escape of air is prevented and the air remaining in the cylinder acts as a cushion that prevents the piston from striking the bottom too violently. If the piston descends too rapidly and strikes the bottom of the cylinder, or "slams," it indicates that so much air escapes that not enough is caught under the piston to cushion it properly; this difficulty can be remedied by reducing the opening of the cock. The lower cylinder is made as nearly air-tight as possible; to permit of the escape of any air that may leak in around the plunger, this cylinder should be provided with a check-valve. The escape of air from the opening in the side of the cylinders makes a hissing sound that may be prevented by connecting the opening to a pipe that leads outside of the engine room. Some dashpots are made with an air chamber that forms a reservoir from which air is drawn by the ascending piston and into which the air is discharged again as the piston descends. This prevents the hissing sound without the necessity of using the pipe.

RELATIVE MOTION OF PISTON, CRANK, AND VALVES.

36. Fig. 11 shows the piston nearing the end of its backward stroke and all the valves closed. The wristplate is in its middle position; hence, the exhaust valves are in precisely the same position, relatively, to their respective ports as would also be the two admission valves were it not that the one *a'* is in the released condition. Observing the arrows on the various valve rods, it will be seen that the exhaust valve *e'* will soon be opened to liberate the steam that is still exerting an expansive pressure in the direction of motion of the piston, while the other valve *e* has but lately been closed; the admission valve *a* will also soon be opened to admit steam against the motion of the piston.

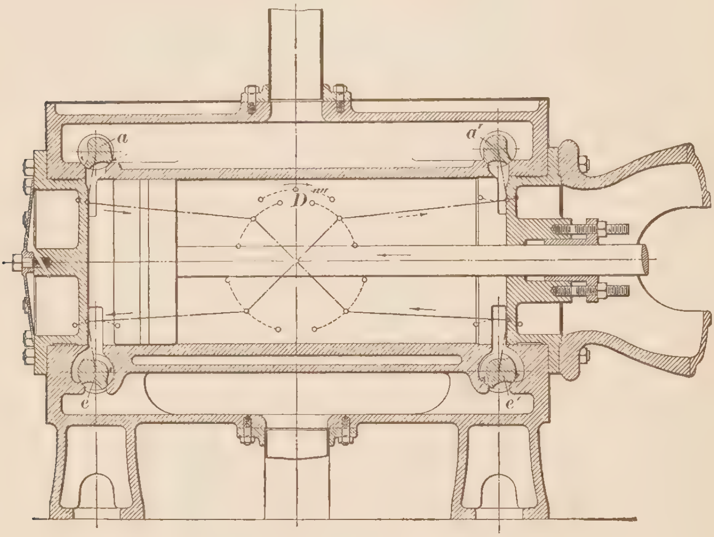


FIG. 11.

37. The diagrams *A*, *B*, *C*, *D*, and *E* of Fig. 12 exhibit the most significant simultaneous positions of the piston and the four valves, together with a skeleton outline of the principal members of the mechanism in their various corresponding positions, during a little more than a complete forward stroke of the engine.

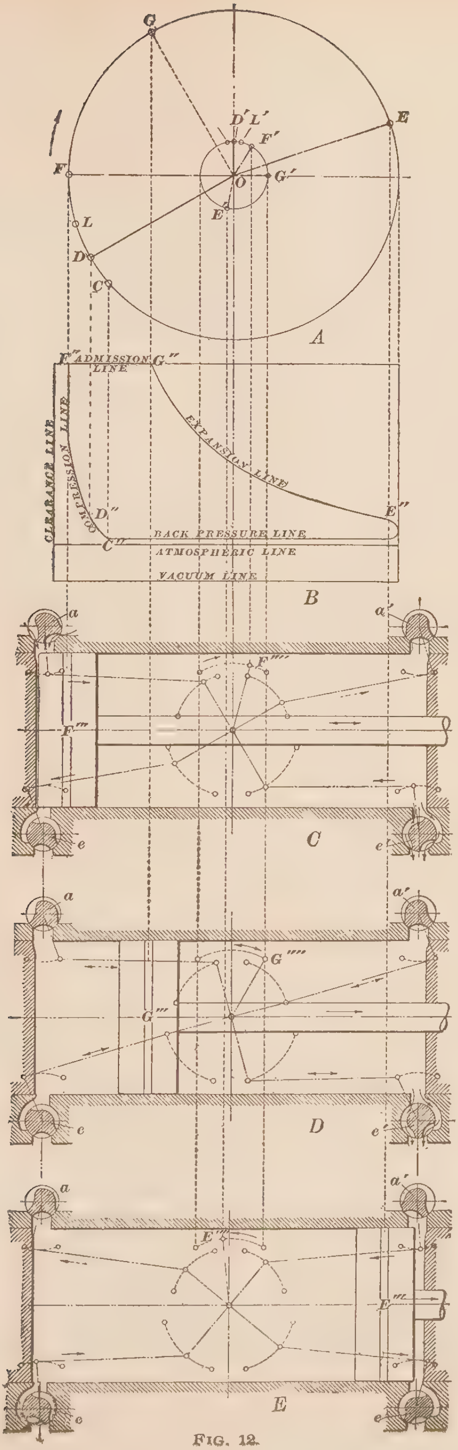

G
E
D′ L′
F′
F
O
G′
E′
L
D
C
A
F″ ADMISSION LINE
G″
CLEARANCE LINE
COMPRESSION LINE
EXPANSION LINE
D″
E″
C″
BACK PRESSURE LINE
ATMOSPHERIC LINE
VACUUM LINE
B
a
a′
F″″
F‴
e
e′
C
a
a′
G″″
G‴
e
e′
D
a
a′
E″″
E‴
e
e
E

FIG. 12.

In all the diagrams the directions of motion of the piston, wristplate, and valve rods are indicated by arrows, a double-headed arrow being shown when a member is in the position in which its direction of motion is being reversed, and a dotted arrow being shown when an admission valve rod is moving without affecting its valve, on account of their connecting parts being in the released condition.

At *A*, in Fig. 12, the respective positions of the crank and eccentric that correspond to the positions in Fig. 11 are *D* and *D'*, and the position *C* of the crank indicates when the closing of the exhaust valve *e* actually took place, and the resulting compression commenced, as indicated at *C''* on the diagram *B*, Fig. 12, while the position *L* indicates where the admission valve *a* will open—in other words, the lead position of the crank. Diagram *C*, Fig. 12, shows the piston at the end of its backward stroke; or, what is the same thing, at the beginning of its forward stroke. By this time both the exhaust valve *e'* and the admission valve *a* have been opened considerably, without, however, reaching the limits of their opening positions, while the exhaust valve *e* has nearly reached the limit of its closing position, and this because the four points of attachment of the valve rods to the wristplate are, as will be seen, so located that the angular closing movements of the valves are very small compared with their angular opening movements.

38. At *D*, in Fig. 12, the wristplate is shown in the position in which its motion is just being reversed by the eccentric, the valve rods being, in consequence, also in the positions in which their motions are reversed, as indicated by the double-headed arrows. According to what we have already learned, this constitutes the limiting position for the releasing of the admission valve *a*, which for that reason is supposed to have just occurred, as indicated by its closed position. As indicated at *G''*, on diagram *B*, Fig. 12, this is then the moment at which in this particular case the expansive reduction of the driving pressure on the piston commences. The exhaust valve *e* has at the same time reached the actual limit of its closing position, while *e′* has reached the limit of its opening position. In this position of its valve rod the admission valve *a′* is picked up, as indicated by one-half of the arrows being represented in full.

G and *G′*, in diagram *A*, Fig. 12, are, respectively, the positions of the crank and the eccentric that correspond to diagram *D*, Fig. 12.

39. In diagram *E*, Fig. 12, the piston is finally represented in that position, near the end of its forward stroke, for which the exhaust valve *e* just begins to open for the liberation of the expanded steam, *e′* having been closed some time previously to produce compression, and the admission valve *a′* also nearing its opening point for the lead position of the piston, which will be reached presently.

Point *E″*, in diagram *B*, Fig. 12, shows the release of the expanded steam due to the opening of *e*, and points *E* and *E′*, in diagram *A*, Fig. 12, represent, respectively, the positions of crank and eccentric corresponding to diagram *E*, Fig. 12.

SETTING CORLISS VALVES.

40. Necessity of Lead and Lap.—In any valve gear the motion of which is derived from the action of an eccentric set at 90° ahead of the crank, the crank will be on its center when the eccentric is at half throw and the eccentric will arrive at its greatest throw, and the opening motion of the valve will cease when the crank is at half stroke. With releasing gears, like the Corliss, the releasing mechanism must operate, if at all, before this point in the eccentric travel is reached; otherwise, the valve will begin to close positively, and at a speed governed by the eccentric on its return stroke. The Corliss valve gear, when properly constructed, closes the valve positively before the end of the stroke, in case the trip does not act or the dashpot fails to close it; this fact does not seem to be generally known. It follows, then, that when the releasing mechanism does not operate, a steam valve set edge to edge with the port opening is closed at the moment the piston reaches the end of its stroke. The exhaust valve on that end of the cylinder opens at the same moment, and there is a tendency to blow through. On the other end of the cylinder the exhaust valve closes and the steam valve opens, also at precisely the same moment, giving the steam another chance to blow through. It is essential, then, that the steam valve shall have a definite advance in its closing movements, relative to the opening of the exhaust valve, in order that it may have a safe working lap before the exhaust port is opened, and that it may not open until after the exhaust valve has closed. The exhaust valves must also have an advance relative to the piston movement, in order that there may be prompt release, and that they shall close on the end towards which the piston is moving before it arrives at the end of its stroke. This is necessary to produce sufficient compression to check the motion of the reciprocating parts of the engine.

41. Effects of Lead.—By setting the eccentric at an angle of more than 90° ahead of the crank, we get an earlier opening and an earlier closing of all the valves, relative to the piston motion. Unfortunately, however, the same amount of lead is not wanted for all the valves, more lead being required on the exhaust valves so that they will close early and thus give sufficient compression. But this does not help us to get the steam valve closed previous to the opening of the exhaust valve nor the exhaust closed before admission of steam. The effect, then, of lead, as derived from the advance of the eccentric to an angle of more than 90° from the crank, is to hasten both the opening and the closing of all the valves, as regards the motion of the piston; therefore, it hastens cut-off and limits its range.

42. Amount of Lap Required.—The effect of lap on a valve is to hasten its closing and retard its opening. Here, again, cut-off is hastened and its range limited. From this it will be seen that when a single wristplate is used there are conflicting conditions in the setting of valves between which a compromise must be effected. No definite rule can be given by which the amount of lap for a valve can be determined. It depends somewhat on the design of the valve and its relative proportions; also on the conditions under which the engine is to work. In all cases the lap increases with the size of the cylinder. The table given below furnishes a fairly reliable guide as to the amount of lap to be given to valves on different sizes of engines:

STEAM LAP AND EXHAUST OPENING FOR CORLISS ENGINES.

Diameter of Cylinder. Inches.	Lap for Steam Valve. Inches.	Exhaust-Valve Opening. Inches.	Diameter of Cylinder. Inches.	Lap for Steam Valve. Inches.	Exhaust-Valve Opening. Inches.
12	$\frac{1}{4}$	$\frac{1}{32}$	30	$\frac{1}{2}$	$\frac{1}{8}$
14	$\frac{5}{16}$	$\frac{1}{32}$	32	$\frac{1}{2}$	$\frac{1}{8}$
16	$\frac{5}{16}$	$\frac{1}{32}$	34	$\frac{1}{2}$	$\frac{1}{8}$
18	$\frac{3}{8}$	$\frac{1}{16}$	36	$\frac{1}{2}$	$\frac{1}{8}$
20	$\frac{3}{8}$	$\frac{1}{16}$	38	$\frac{9}{16}$	$\frac{3}{16}$
22	$\frac{3}{8}$	$\frac{1}{16}$	40	$\frac{9}{16}$	$\frac{3}{16}$
24	$\frac{7}{16}$	$\frac{3}{32}$	42	$\frac{9}{16}$	$\frac{3}{16}$
26	$\frac{7}{16}$	$\frac{3}{32}$	44	$\frac{5}{8}$	$\frac{1}{4}$
28	$\frac{7}{16}$	$\frac{3}{32}$	46	$\frac{5}{8}$	$\frac{1}{4}$

43. Centering Rocker-Arm and Wristplate.—A method of centering the wristplate is illustrated in Fig. 13 by the plumb line $x y$ let fall from the hand. Usually, however, it is not necessary to resort to this scheme; in most cases there will be found three marks, as at c, b, d, Fig. 14, on the wristplate bracket and another mark a on the hub of the wristplate, by means of which the wristplate may be

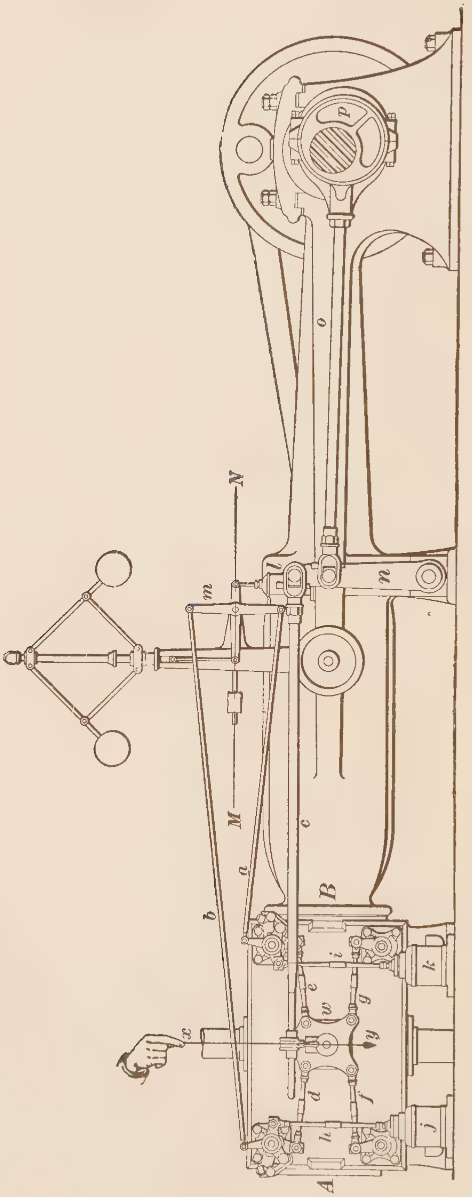


FIG. 13.

centered. The marks are so located that *a* is opposite *b* when the wristplate is at its center of motion. At the two extremes of motion *a* is opposite either *c* or *d*.

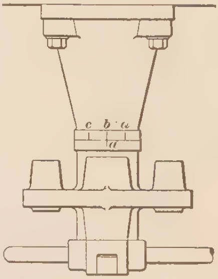


FIG. 14.

It may be well, however, to test these marks, or rather to see that the eccentric and carrier rods have proper adjustment relative to the motion of the wristplate. To do this, rotate the eccentric *p*, Fig. 13, on its shaft, having the eccentric rod *o* connected and the carrier rod *c* hooked over on the wristplate; then notice whether or not the rocker-arm *n* is equidistant in its extreme travel each way from a plumb line let fall through the center of its pin. If it is not, make it so by adjusting the length of the eccentric rod *o*. Next, see if the mark on the wristplate hub agrees with those on the bracket at full throw each way; if not, the remedy is to change the length of the carrier rod *c* until there is perfect agreement. Having tested the marks temporarily, secure the wristplate *w* at its center of motion.

44. Adjusting the Lap.—Upon removing the back bonnets, or caps, from the ends of the valve chambers, so that the rear ends of the valves are exposed, a mark will usually be found on each face of the valve ports, showing the location and width of the port openings in relation to the cylinder. Upon the ends of the valves, marks in line with the opening edges of the valves will also be found. See Fig. 15. Possibly in some of the older types of engines these may be missing; in such a case the valves will have to be removed to locate the port openings and the opening edges of the valves. Consulting the table, Art. **42,** find for the engine being adjusted the lap for the steam valves and the opening to be given the exhaust valves. By lengthening or shortening the rods leading from the wristplate to the valve arms, bring the opening edges of the valves to positions corresponding with the desired lap or opening. While making these adjustments, it is, of course essential that the steam latch shall be hooked on the stud

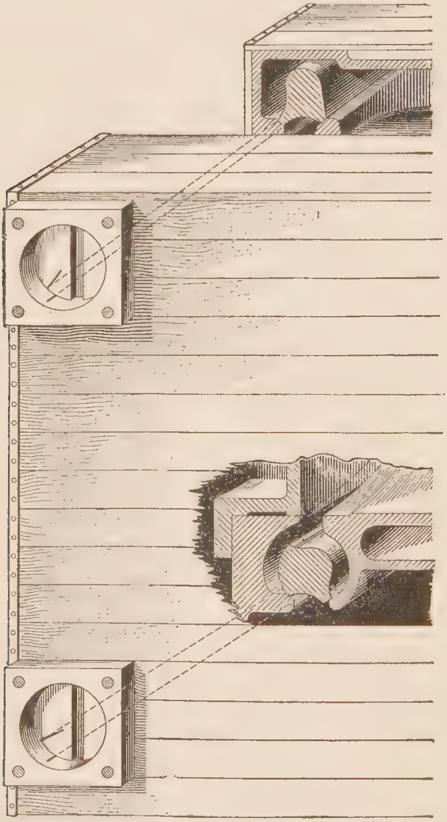


FIG. 15.

If a record is kept of how much the valve moves at one turn of the adjusting nut on the rod, future adjustments may be made without the removal of the bonnets.

45. Adjusting the Lead.—All the valves are now supposed to be in their proper positions when the wristplate is at its center of movements. The section shown in Fig. 11 gives this position, except that *a′* is supposed to be unhooked and to stand in a position of full closure. The next step is to locate the eccentric at the proper angle ahead of the crank to give sufficient lead. First, set the engine exactly on its center; and with the carrier rod hooked on the wristplate stud revolve the eccentric on the shaft in the direction in which the engine is to run, until it is at an angle greater than 90° ahead of the crank or until the steam valve on the end at which the piston stands is just beginning to open. In this position the eccentric must be secured to the shaft. Then turn the engine to the other center and see if the steam valve on that end has the same amount of opening as the other had. It should and will have the same amount if all adjustments have been carefully made.

46. Adjusting the Governor.—For the purpose of adjustment, block the governor so that the balls stand in the position they would assume at normal speed (about mid-position) and fasten the reach-rod lever *m* at right angles to a line *M N*, Fig. 13, midway between the reach rods *a* and *b*. Now turn the engine to the point at which cut-off should occur (usually about $\frac{1}{5}$ stroke) and adjust the reach rod for that end, so the valve will trip at that point. The valve and the reach rod for the other end of the cylinder must be adjusted in a like manner. To determine the point of $\frac{1}{5}$ stroke, mark the length of stroke on the cross-head guides and measure off $\frac{1}{5}$ of this from each end. After a few trials, partially rotating the engine back and forth, at the same time making careful adjustments of the reach rods, cut-off can be made to take place at exactly similar points for each end. It is well, now, to lower the governor to the lowest position and observe that the cut-off mechanism does not work, but allows steam to be taken during the full stroke of the piston.

To regulate the sensitiveness of the governor, vary the amount of opening between the two ends of the dashpot cylinder *l*.

47. Dashpot Adjustments.—Care must be taken, in making adjustments of a valve gear, not to overlook even the smallest detail. For instance, the dashpot rods must have such a length that the steam arm will be in a position where the hook will surely hook on when the dashpot plunger is home.

48. General Summary.—To regulate the point of cut-off so that the same amount of steam is admitted at both ends, adjust the lengths of the reach rods; to give more or less steam lap, lengthen or shorten the steam rods. A change in the exhaust rods likewise affects the cushion and release. After the eccentric has once been properly set, it is not necessary to disturb it in ordinary cases. If the dashpot rod is too short, the latch will not hook. Look out for this. It is an excellent plan to mark every position; one can then tell at a glance if any adjustments have been disturbed. As a final test of the valve setting an indicator diagram is to be recommended.

CORLISS GEARS WITH TWO ECCENTRICS.

49. In order to secure a satisfactory steam distribution when the steam and exhaust valves of a Corliss engine are both driven by the same eccentric, it is necessary to give the eccentric a certain angle of advance. It has been explained previously that the releasing gear cannot act on the valves after the eccentric has passed its extreme throw position. The result of these conditions is that with a single eccentric the releasing gear cannot be made to act after the piston has passed through a fraction of its stroke always less than ½ and often little more than ⅜. If the load on the engine is so heavy that cut-off does not take place before this fraction of the stroke has been completed, the steam valves will not be under the control of the governor, but will be closed only by the action of the eccentric.

The range at which cut-off can be controlled by the governor can be considerably extended by the use of two eccentrics and wristplates—one for the steam valves and one for the exhaust. The exhaust eccentric can then be given the angular advance required by the conditions, while the steam valves may be made to open early enough to secure the desired lead with the steam eccentric set without any angular advance or even at an angle of less than 90° with the crank. The steam eccentric thus reaches its extreme throw position later in the stroke, and the range within which cut-off is controlled by the governor is correspondingly lengthened.

50. Amount to Which the Range of Cut-Off Can Be Economically Extended.—By setting the steam eccentric at a small enough angle with the crank, the range of cut-off with a double eccentric might be extended to nearly or quite full stroke. Such a range, however, is not desirable; it is not economical to use an engine under a load that makes it necessary for steam to follow the piston at boiler pressure for such a large part of the stroke, and, in addition, an attempt to secure a long range of cut-off by setting the eccentric back near to the crank position will cause the steam valves to open very slowly at the beginning of the stroke, thus producing wiredrawing of the steam and reducing the economy when working under normal loads. Satisfactory results can be obtained with the eccentric set at an angle of as little as 81° with the crank, 9° back of the 90° position; this provides for an extreme range of cut-off of about $\frac{7}{10}$ stroke.

51. In designing and adjusting the cut-off gear with a double eccentric motion, it is important to make sure that the steam valve for each end of the cylinder will always be closed by the action of the releasing mechanism before the exhaust valve for that end opens. If this precaution is not observed, there will be danger that with a heavy load the steam and exhaust valves for one end of the cylinder will both be open at the same time, thus allowing steam to blow from the steam chest directly through the cylinder into the exhaust.

SAFETY STOPS.

52. Most, if not all, modern Corliss valve gears are fitted with some kind of a safety stop that will prevent the running away of the engine in case the governor belt should break. In that case the governor will slow down and then stop, and hence, if no safety device is fitted, act as if the engine had slowed down; that is, it will lengthen the cut-off so as to speed up the engine, thus allowing it to run away. To prevent this, some kind of a device is usually fitted that prevents the steam valves from hooking on when the governor stops. When the engine is about to be started, this safety device would prevent the starting of the engine, as the steam valves cannot operate, and hence a collar is usually fitted to the governor by means of which it can be raised sufficiently to allow the steam valves to hook on in starting. As soon as the engine is running this collar obviously must be lowered again so as to render the safety stop operative.

OSCILLATING VALVES WITH POSITIVE-MOTION GEARS.

53. In order to secure the advantages of the four oscillating valves and the wristplate motion of the Corliss gear without the attendant disadvantages of the restricted speed of revolution that is imposed by the releasing gear, and also to secure the positive valve motion and the superior degree of regulation that can be obtained with a shaft governor, a number of engines have been designed with the Corliss valves and wristplate driven by a shifting eccentric controlled by a shaft governor. Since it is often desirable to keep the release and compression constant, some of these engines use two eccentrics. One of the eccentrics drives the steam valves; it is controlled by the governor so as to regulate the amount of valve opening and the point of cut-off in practically the same manner as with a slide-valve engine. The other eccentric is fixed on the shaft and drives the exhaust valves. A variable cut-off with constant release and compression are thus obtained.

54. Engines of this class are often called **high-speed Corliss** engines. The general instructions for centering the wristplate and rocker-arms and for adjusting the valves for lap and lead that were given in Arts. **43** to **45** for the Corliss gear apply to engines with positive-motion oscillating valves. In most cases, however, it will be necessary to give the steam valves an amount of lap considerably in excess of that called for in the table in Art. **42**.

STEAM TURBINES

CONSTRUCTION AND OPERATION

GENERAL INFORMATION

INTRODUCTION

1. The translation of the energy in steam into rotary motion by means of mechanical appliances has been attempted at various times and places for the past two thousand years. The earliest form of steam engine was naturally the one that required the least mechanical skill and precision of workmanship, and took the form of a globe mounted on trunnions, with two orifices for the discharge of steam, situated at diametrically opposite points of a circle on the sphere drawn midway between the points of suspension. These orifices were so fitted with bent tubing that they discharged the steam in a direction opposite to the direction of rotation of the sphere, and tangential to the circle in which they moved.

The rotation of the sphere on its trunnions is caused by the reaction of the jet of steam as it escapes into the surrounding atmosphere, and hence it is a **reaction motor.** Its principle of operation may be explained as follows: The pressure at every point of the inner surface of the bent discharge tube is balanced by the pressure on a point

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opposite; hence, if there were no opening for the escape of steam, there would be no tendency to rotate. But, by making an opening in the end of the bent tube the pressure on the surface opposite becomes unbalanced and rotation begins. From this it follows that the force tending to rotate the sphere is proportional to the area of the opening, the steam pressure remaining constant. An engine built in England about 1883 on the model of this ancient motor showed a water consumption of about 40 pounds per brake horsepower.

The next appearance of a steam motor in history was in 1629, when Branca proposed a type of motor in which the steam upon issuing from a nozzle impinged upon the blades, or vanes, of a paddle wheel. This type may be regarded as the forerunner of the **impact turbine,** of which the De Laval turbine is the present representative. The Parsons turbine, differing from the De Laval machine mainly in the method of securing a high ratio of expansion of the steam, dates from 1884. It is claimed by the inventor that it is a combination of the reaction and impact types. The Curtis turbine was placed on the market in 1903 by the General Electric Company, Schenectady, New York. The De Laval, Parsons, and Curtis turbines are the only types that have at present attained any degree of commercial success for general work in the United States of America.

GENERAL ADVANTAGES

2. The advantages claimed for the De Laval steam turbine are no leakage from wear, small friction loss, high efficiency with variable loads, no moving parts under pressure, close speed regulation, simplicity of construction, perfect balance, small foundations, small space occupied, ease of erection, automatic oiling, no danger from water, and long life.

The advantages over reciprocating engines claimed for the Westinghouse-Parsons turbine are increased speed greater economy of steam, reduced weight, reduced space occupied by machinery, reduced cost of attendance, reduced repair bills, and reduced vibration. The same advantages are claimed for the Curtis turbine, comparing it with reciprocating engines.

DE LAVAL TURBINE

WHEEL AND NOZZLES

3. The steam turbine as produced by De Laval was first made a success in 1883. Its first application was in connection with dairies in the cream-separator machines, where its high speed, simplicity of construction, and compactness made it an almost ideal motor for this class of work, and hence it was extensively adopted for it.

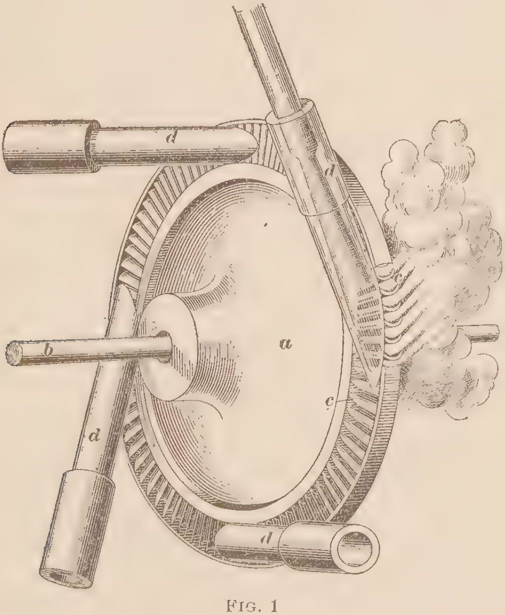

FIG. 1

In the form in which it was first placed on the market, it did not utilize the expansive capabilities of the steam in producing a high velocity of discharge, and, consequently, the turbine was uneconomical in the use of steam. Means for utilizing the steam expansively were applied later, however, and made the turbine quite an economical machine.

4. The De Laval turbine consists essentially of a wheel *a*, Fig. 1, suitably mounted on a shaft *b*, free to revolve in bearings. The wheel carries at its periphery a large number of radial vanes, or blades, *c*, *c* of a curved cross-section. In order to show these vanes clearly, a piece of the rim has been broken away. Steam is led to the wheel by means of stationary nozzles, as *d*, *d*, varying in number from 1 to 12, according to the size of the turbine. The nozzles are so placed that the steam issuing from them at a high velocity impinges on the blades, and thus rotates the wheel, which is enclosed in a suitable casing. Proper means are provided for regulating the quantity of steam discharged by the nozzles, and thus the horsepower developed is varied.

GENERAL DESCRIPTION

5. The general arrangement of the component parts of the De Laval steam turbine is shown in the sectional plan view given in Fig. 2. The turbine wheel *a* is mounted on the flexible shaft *b* running in the three bearings *c*, *d*, *d* and the so-called flexible bearing *c'*. The bearings *d*, *d* are rigid; the bearing *c*, however, is fitted in such a manner that it can accommodate itself to any bending of the shaft. The flexible bearing *c'* is really not a bearing at all, since it does not support the shaft, but merely closes the opening where the shaft passes out of the casing. In reality it is a stuffingbox so mounted as to accommodate itself to any bending of the shaft, and preventing the entrance of air to the casing *e* when running condensing, and the escape of the steam when running non-condensing. The turbine wheel runs within the casing *e* that carries the nozzles that owing to the view taken cannot be shown. The position of the nozzles in regard to the wheel has, however, been clearly illustrated in Fig. 1. The turbine wheel has such a high rotative speed that very little machinery in use can be coupled directly to the shaft *b*; in order to reduce this high rotative speed to the limits of usual practice, gearing is resorted to. The shaft *b* is extended as shown and carries the two pinions *f*, *f* meshing with the gear-wheels *g*, *g*, which generally have 10 times as many teeth as the pinions, thus reducing the speed to $\frac{1}{10}$ the speed of the turbine wheel. The gears have helical teeth, and one set of gears has right-handed helical teeth while the other set has left-handed helical teeth. In consequence of this, there is practically no chance for the shaft to move endwise. The gears are enclosed in a closed case partially filled with oil. The machine to be driven by the turbine is connected to the shaft *h*, either by being coupled to it directly, or by being belted to a pulley keyed to the shaft. The shaft *h* carries on one end the centrifugal governor *i* that operates a double-seated throttle valve by means of a bell-crank lever. This lever *l* is clearly shown in Fig. 3, which is an elevation of the same turbine shown in Fig. 2,

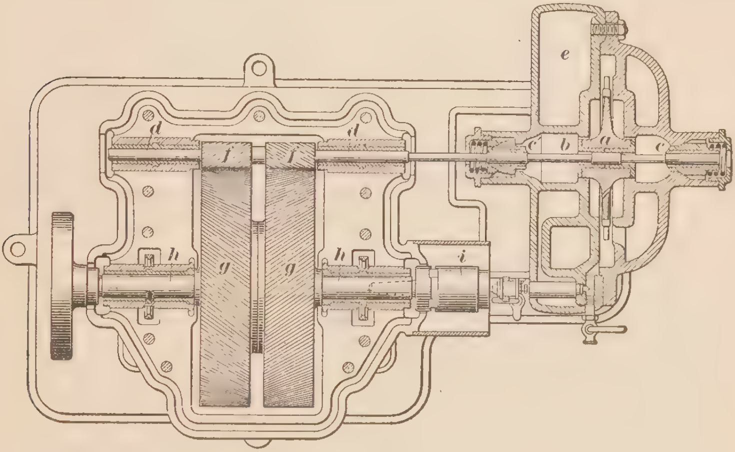

FIG. 2

and where all visible parts have been lettered the same as in Fig. 2. Steam is admitted to the turbine through a pipe screwed into the flange *m*; the exhaust pipe leading either to the atmosphere or to a condenser is bolted to the flange *n*.

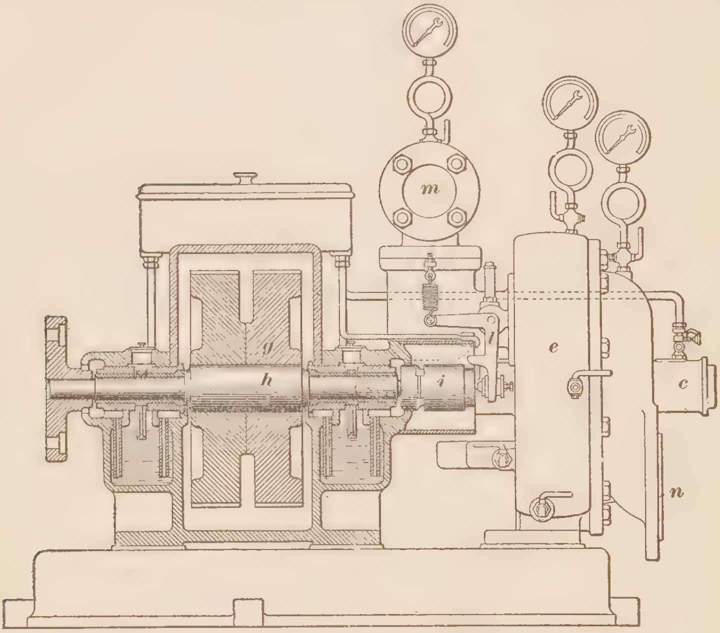

FIG. 3

GOVERNOR AND VACUUM VALVE

6. The construction of the **governor** is shown in Fig. 4. It consists of a frame fastened to the end of the shaft by a taper shank and carrying the weights *a*, *a* hung on knife-edge bearings. These weights fly outwards under the action of the centrifugal force until the resistance of the springs *b*, *b* is equal to the centrifugal force. In moving outwards, the weights force the governor stem *c* to the right, which in turn rotates the bell-crank and causes the governor valve operated by it to assume a position corresponding to the speed of the governor. When the speed falls, the governor weights move inwards, and the governor valve is opened farther by them, thus admitting more steam. When the speed rises, the centrifugal force increases and the governor weights move farther outwards; this closes the governor valve farther and thus reduces the amount of steam admitted.

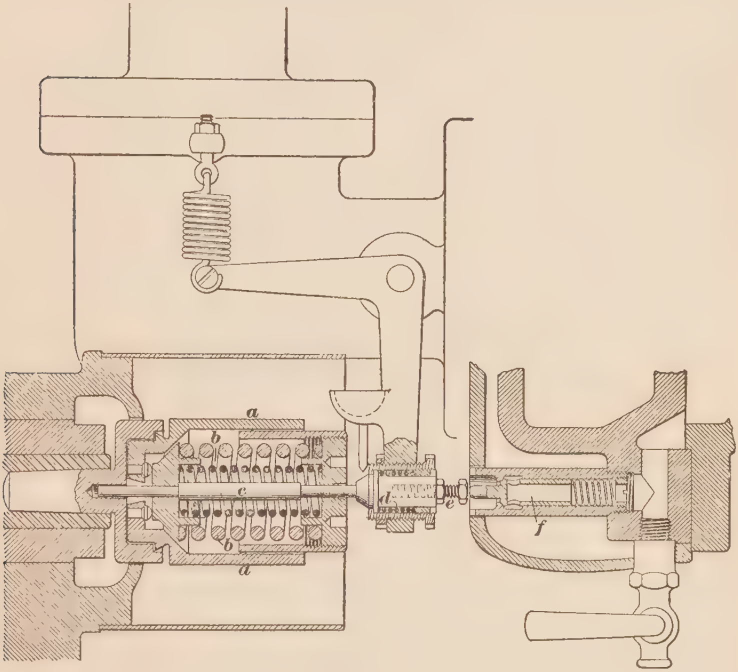

FIG. 4

The speed at which the turbine will run can be varied somewhat by changing the tension of the governor spring, increasing the tension to make the turbine run faster and decreasing it to make the turbine run slower.

7. When the turbine is running condensing, it has been found that the governor valve alone will not give a sufficiently close regulation during a sudden and excessive decrease of load. To assist the governor valve in controlling the speed under these conditions, the **vacuum valve** has been designed. Its function is to admit air to the space in which the wheel revolves and thus to destroy or impair the vacuum; the resistance of the air then effectually checks the speed of the wheel. The vacuum valve is operated by the governor as follows: When the speed under a sudden decrease of load becomes excessive, the governor first closes the governor valve entirely. The governor weights continue to move outwards after this and force the governor stem *c*, Fig. 4, farther to the right, compressing the spring *d* in doing so. The adjustable stud *e* then strikes the end of the vacuum valve *f* and forces this valve inwards, thus admitting air. When the speed has been sufficiently reduced, the governor weights move inwards and consequently the stud *e* moves away from the vacuum valve, allowing it to close. As the weights continue to move inwards, they open the governor valve again and admit steam to the turbine once more.

PURPOSE OF THE FLEXIBLE SHAFT

8. The vibration of the shaft and wheel at the high speeds is reduced to a minimum by an extremely careful balancing of the revolving parts, and by making the shaft itself slender and hence flexible. It was found long ago that, with rigid shafts running in rigid bearings, the most careful balancing failed to prevent excessive vibration at very high speeds, as the centrifugal force generated by the absence of absolute balance was very large. The discovery was made, however, that by either making the shaft slender and hence flexible, or by making it very rigid and the bearings capable of lateral accommodation, the rotating parts would vibrate until the so-called *critical speed* was reached and that after passing this speed the vibration would gradually cease. The exact reason for this action is at present a matter of speculation; the most commonly accepted hypothesis is that the special construction permits the parts to revolve about their axis of gravity instead of their geometrical axis, the axis of gravity being the line joining the centers of gravity of the different parallel sections at right angles to the shaft into which the revolving parts may be conceived to be divided. This discovery has been employed in the De Laval turbine, making the shaft small and hence flexible.

DE LAVAL NOZZLE

9. The means by which steam is used expansively in the De Laval turbine is one of the distinguishing characteristics of this type of turbine, and is known as the **De Laval nozzle.** It consists of a tube *a*, Fig. 5, with a contracted opening, or *throat*, near the inlet end. The steam passage in the nozzle diverges from the throat to the outlet end. The inlet ends of the nozzles are within the annular passage *b* concentric with the shaft, and to which the live steam is admitted. The valve *c* regulates the amount of steam entering each nozzle, and is also used for cutting one or more nozzles out of service when the turbine is not to be worked under a full load.

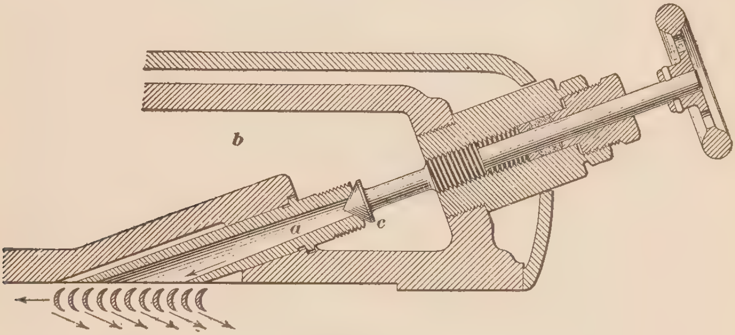

FIG. 5

The purpose of the expansion nozzle is to greatly increase the velocity with which the steam strikes the blades of the turbine wheel. Since the amount of work a moving body can do in being brought to rest varies as the square of the velocity, it can readily be seen that the work a given weight of steam can do in a turbine is greatly increased by even a small increase in its velocity.

It may appear on first thought that the velocity of steam issuing from an orifice, as a nozzle, can be increased best by increasing the difference in pressure between that of the steam entering the nozzle and that in the space into which the steam is discharged. In carrying out this idea in practice, however, the fact is encountered that the velocity of steam issuing from an orifice in a thin plate, or from a converging nozzle, into a space of less pressure only continues to increase as the difference in the pressure increases until the pressure in the space of less pressure is approximately $\frac{4}{7}$ of the steam pressure at the inlet side. After this point is reached, the velocity of the steam remains practically constant, no matter how much farther the difference in pressure is increased.

From the foregoing statements it will be seen that there is a limit beyond which it is inadvisable to raise the pressure of the entering steam, since no increase of velocity will result from a further increase, considering an orifice in a thin plate or a converging nozzle.

It has been discovered that by discharging the steam through a nozzle with diverging sides, as the De Laval nozzle illustrated, an increase of velocity at the discharge end beyond the limit imposed by the phenomenon explained above can be obtained, the steam in passing through the nozzle expanding down to a low pressure. The object, then, of the De Laval nozzle is to impart a high velocity to the steam, thus giving a great kinetic energy.

WESTINGHOUSE-PARSONS TURBINE

ACTION OF STEAM

10. The steam motor known to the engineering profession by the name of the Parsons turbine is the invention of an English engineer, Charles Parsons, and was quite highly developed before being introduced into America. The first Parsons turbine was built in 1884 and supplied power for the manufacture of electrical apparatus in Gateshead, England. The Westinghouse Machine Company, of

Pittsburg, Pennsylvania, have obtained the right to manufacture the Parsons turbine in this country, and it is now placed on the market under the name of **Westinghouse-Parsons steam turbine.**

11. In the Westinghouse-Parsons steam turbine the steam is discharged successively upon a number of wheels carrying blades, and placed side by side on the shaft, fixed guide blades being placed between each pair of wheels. This turbine differs radically from the De Laval turbine in the manner in which expansion is secured. In the De Laval turbine the expansion takes place in the nozzle and the steam strikes the wheel at a very low pressure; in the Westinghouse-Parsons turbine expansion takes place as the steam passes from one wheel to the other, and steam at full boiler pressure impinges on the first wheel, the pressure decreasing as the steam passes from wheel to wheel.

The relative arrangement and shape of the fixed guide blades and the blades in the wheels is shown in Fig. 6, where the line *A B* represents the axis of rotation. The steam on entering the turbine casing flows in a general direction parallel to the shaft, entering first the curved guide blades *a* that deflect it against the blades of the first wheel *b*. The impact of the steam causes the wheel to revolve. The steam rebounds from the moving blades to the guide blades *c* that deflect it again against the moving blades of the wheel *d*, whence it passes to another set of guide blades, and so on. It will be understood that all of the turbine wheels are fastened to the same shaft and revolve together. The wheels are caused to rotate both by the impact of the steam delivered against their blades, and the reaction of the steam rebounding from them. Hence, it is seen that the

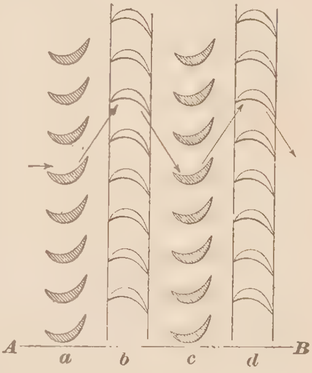

FIG. 6

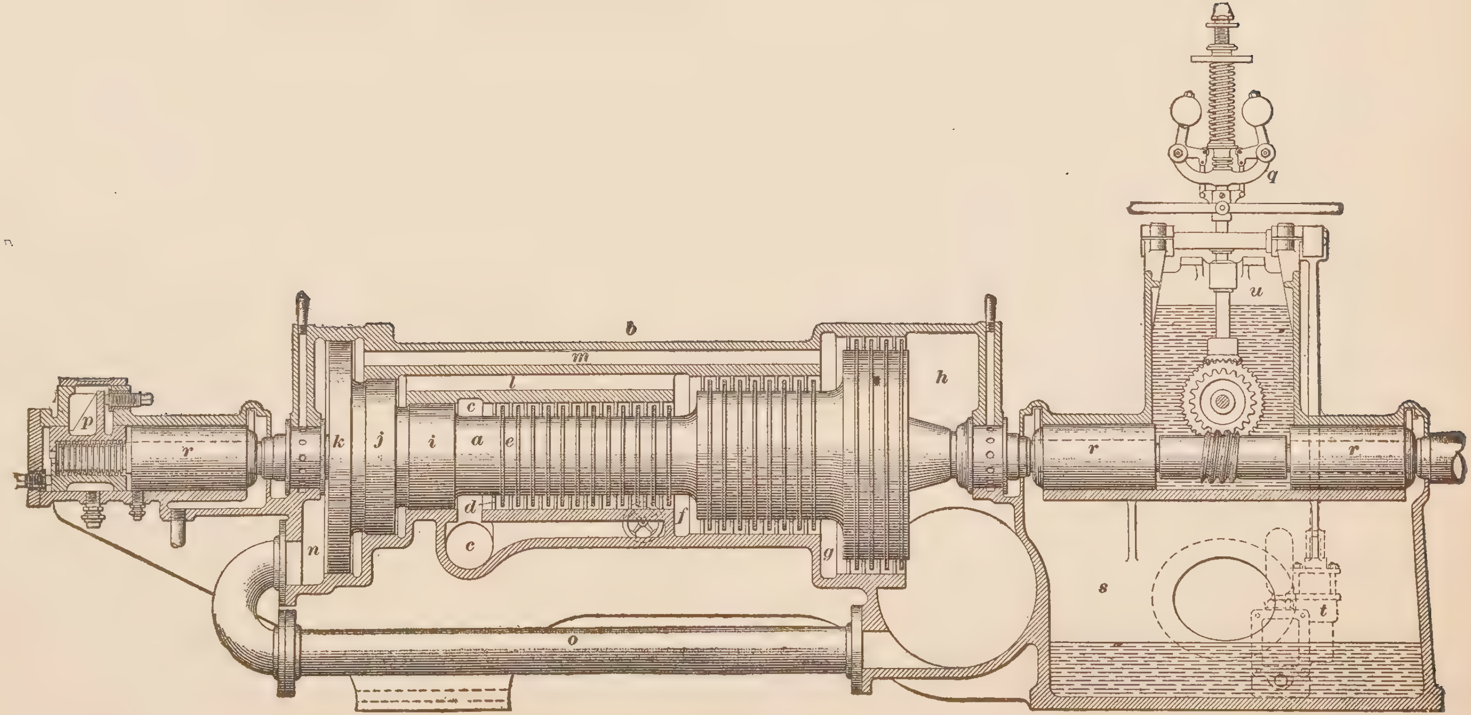

FIG. 7

Westinghouse-Parsons steam turbine is a combination of the reaction and impact types of the turbine.

12. The steam yields the kinetic energy due to its velocity of flow in the following manner: It enters the first stationary set of guide blades at its maximum velocity and is discharged against the moving blades, from which it rebounds at a reduced velocity and enters the space containing the second set of guide blades. This space is larger than the first space containing guide blades, and in consequence the steam expands. In doing so its velocity of departure is increased somewhat above the velocity with which it entered the second space, but not up to the velocity with which it entered the first space containing guide blades. The steam leaves each successive space containing guide blades at a velocity below that with which it departed from the preceding space, until on finally leaving the turbine its velocity is very low. From the foregoing it will be seen that the steam does not part all at once with the kinetic energy it is practicable to abstract, as occurs in the De Laval turbine, but parts with it in successive stages. In the De Laval turbine, in order to get maximum efficiency, the velocity of the wheel must be about 47 per cent. of the velocity of the steam, and as this latter velocity is very high, about 4,000 feet per second, the turbine wheel must have a very high rotative speed and the motor requires to be geared down for most work. In the Westinghouse-Parsons turbine, however, the velocity is reduced gradually, and consequently the peripheral speed, that is, the speed of rotation, of the wheels is comparatively low. This is a decided advantage for many purposes, since it allows the motor to be connected directly to many machines.

GENERAL ARRANGEMENT

13. The general arrangement of the parts of a Westinghouse-Parsons steam turbine is given in the sectional view shown in Fig. 7. The turbine shaft *a*, carrying three sets of moving blades, rotates in suitable bearings and within the casing *b*. Steam at boiler pressure enters the annular passage *c*, whence it passes to the first set of stationary guide blades *d*, which direct it against the blades of the wheel *e*. After passing the first set of wheels, the steam, which now has lost considerable of its velocity, flows into the annular chamber *f* and impinges on the second set of wheels, there parting with more of its kinetic energy. The steam passes then into the annular chamber *g* and impinges on the third set of wheels, finally passing at a very low velocity into the chamber *h*, and thence either to the atmosphere or to a condenser. In order to counteract the end thrust of the steam against the moving blades, the shaft is fitted with balance pistons *i*, *j*, and *k*. The piston *i* has an area equal to that of the moving blades of the first set of wheels, and as the steam pressure acts against this piston in a direction opposite to that in which it acts against the blades, the end thrust is balanced. Likewise, the end thrust against the second and third set of blades is balanced by the pistons *j* and *k*, which are subjected to the steam pressure existing in the chambers *f* and *g*, since the chambers in which these pistons revolve are connected by the passages *l* and *m* to the chambers *f* and *g*. In order that the pressures on the outer side of the balance piston *k* and the outer side of the largest set of blades may be equal, the chambers *h* and *n* are connected by a pipe *o*. An adjustable bearing *p* similar to a thrust block confines the turbine shaft longitudinally, and at the same time resists any end thrust that may not be balanced by the balance pistons. A governor *q* serves to regulate the speed of the engine. Live steam can be admitted to the chamber *f* by a by-pass valve for the purpose of increasing the power of the turbine. This, however, is accompanied by a reduction in the economy. Flexible bearings *r*, *r*, *r* permit the shaft and wheels to revolve about their axis of gravity. All bearings are lubricated by oil under pressure, which flows back by gravity to the reservoir *s*, whence the pump *t* forces it back into the supply reservoir *u*, from where it passes once more to all bearings.

FLEXIBLE BEARINGS

14. The means adopted for preventing vibration are an extremely careful balancing of the revolving parts combined with a construction permitting a lateral accommodation of the bearings. In the Westinghouse-Parsons steam turbine the shaft is made just as rigid as possible.

The construction of the bearings is as follows: The gun-metal sleeve forming the bearing proper is prevented from rotating by means of a dowel. This sleeve is surrounded by three loosely fitting tubes. The space between the tubes themselves and the sleeve is filled with oil, which is supplied under pressure. Owing to the oil being a yielding medium, the sleeve can move laterally to accommodate itself to a displacement of the geometrical axis of the shaft.

GOVERNOR

15. The governor, as an inspection of Fig. 7 will show, is a spring-loaded fly-ball governor. By adjusting the tension of the spring loading it, the speed of the turbine can be varied within small limits. The governor valve controlling the admission of the steam to the turbine is not attached directly to the governor, but is operated by steam admitted to its power cylinder and exhausted therefrom by a pilot valve controlled by the governor.

The governing mechanism is shown in Fig. 8. The steam-admission or governor valve is attached to a stem *a* carrying a piston *b* working in a cylinder *c*. A spring *d* placed on top of the piston *b* tends to return it to, and keep it in, its lowest position, where the admission valve is fully closed. In operation, steam at boiler pressure leaks past the stem *a* and raises *b*, thereby opening the governor valve. Steam is exhausted from the under side of the piston by means of the pilot valve *e*, thereby lowering the pressure below *b* and thus permitting the spring *d* to close the governor valve. The pilot valve is given a to-and-fro motion while the turbine is running by means of an eccentric on the shaft driving the governor, the eccentric rod *f* being attached to the lever *g*, which in turn is connected to the lever *h* by the link *i*. The lever *h* in turn is connected to a crank *j* by a link *k*, the crank *j* being connected to the pilot valve *e*. The fulcrum of the lever *h* is placed on the governor collar *l*, and consequently under any change of speed, since the governor balls *m*, *m* move in or out and raise or depress the collar, the height of the fulcrum changes.

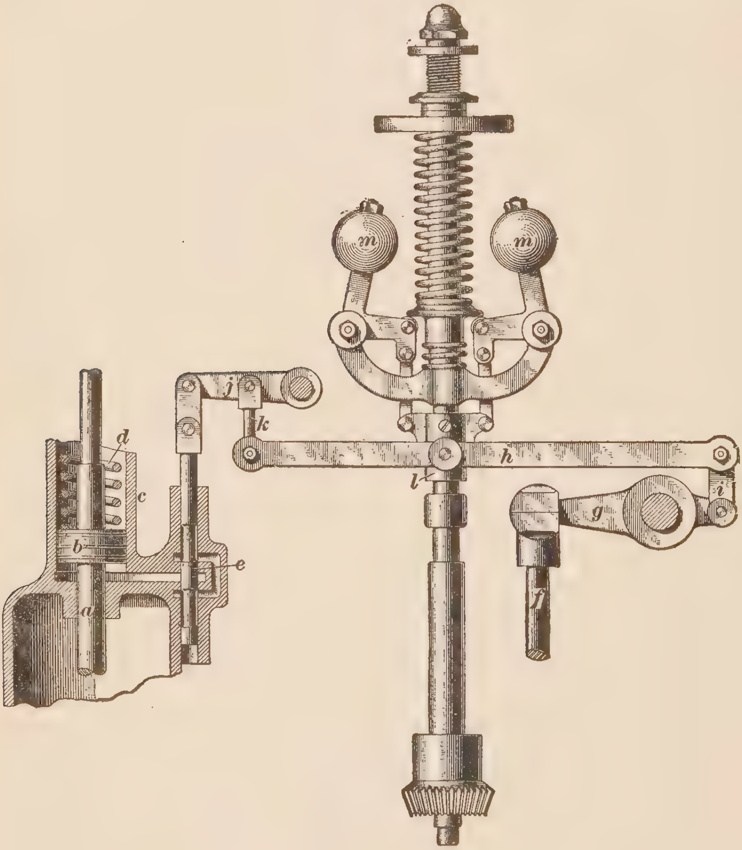

FIG. 8

16. The action of the governor can best be studied by considering first of all the action of the admission valve while the turbine is running at a constant speed. With each revolution of the shaft when it is driving the governor, the pilot valve is opened and the steam is exhausted from the under side of the piston shown at *b*, which moves downward and causes the steam-admission valve to close, thus completely closing the steam inlet. After a certain interval the pilot valve closes the exhaust port from the under side of the piston, and the accumulation of pressure opens the admission valve again, admitting steam once more to the turbine. This alternate closing and opening of the admission valve occurs about 150 times per minute. It is thus seen that the steam is admitted in puffs, as it were.

Reflection will show that the volume of steam admitted at each opening of the governor valve varies directly as the length of time the valve remains open. The object of the governor is to regulate this length of time, which is accomplished by causing the pilot valve to make its strokes in a higher or lower position, according to which way the speed varies, thus changing the length of time the port leading below the piston *b* remains open. Let us suppose the speed of the turbine increases. Then, the governor balls *m*, *m* fly outwards, and in doing so pull up the governor collar *l*. As a result of this, the pilot valve is also lifted and now makes its up-and-down strokes in a higher position. In consequence of this, the port leading below the piston *b* remains open for a greater length of time than before, with the result that the spring *d* is permitted to hold the steam-admission valve closed longer, or in other words, shortens the time during which steam is admitted to the turbine. The result of this decrease in the volume of steam admitted to the turbine is a reduction of its speed down to the normal speed. When the speed drops below that of the normal speed, the pilot valve drops to a lower position, thus reducing the time during which steam is exhausted from below the piston *b*, and consequently increasing the time the steam-admission valve remains open. The result is that more steam is admitted at each opening of the valve, which continues until the speed has increased to the normal speed.

The object of making the admission valve act intermittently, as described, is to admit the steam to the turbine at full boiler pressure irrespective of any variation in load. With a full load the puffs of steam admitted will be almost continuous.

CURTIS TURBINE

ARRANGEMENT OF BLADES AND NOZZLES

17. The Curtis steam turbine, which is manufactured by the General Electric Company, of Schenectady, New York, in some respects combines the principles which are employed in the operation of the De Laval and Parsons turbines, using the expansion nozzles of the first make and the alternate rows of stationary and moving blades acted upon by the steam that are employed in the second make of steam turbine mentioned. The Curtis turbine, as manufactured at present, is vertical; that is, the shaft is vertical and the wheels carrying the moving blades revolve in horizontal planes.

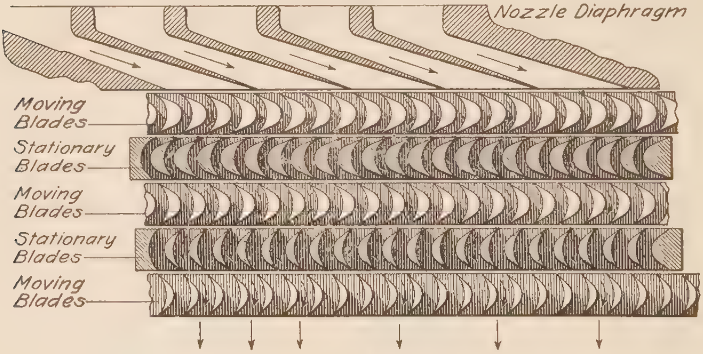

FIG. 9

18. A diagrammatic view showing the arrangement of the expansion nozzles, stationary blades, and moving blades is given in Fig. 9, the direction of flow of the steam being indicated by the arrows. The steam is first expanded to a low pressure in the expansion nozzles, having its velocity of flow increased thereby; the kinetic energy of the steam is then abstracted in successive stages by the rows of moving blades. A comparatively low speed of rotation is secured by making the wheels carrying the moving blades large in diameter; this allows the speed at the circumference to be high enough to suit the velocity of flow of the steam.

ARRANGEMENT OF TURBINE

19. A sectional view of one form of the Curtis turbine is given in Fig. 10, the particular machine shown being designed to normally develop 6,700 actual horsepower. The vertical shaft *a* is supported sidewise by bearings *b*, *b* and in a vertical direction by a step bearing *c*; the shaft carries a number of horizontal wheels, as *d*, *d*, two in this case and three or four in others. These wheels at their circumference carry steel rings in which the blades are formed by cutting from the solid metal; stationary guide blades, as *e*, *e*, are placed between each two rings of moving blades, as *f*, *f*. The stationary blade rings are supported by the casing *g*. Governing is affected by valves, as *h*, placed in the passage leading to each expansion nozzle, these valves being operated by a governor not shown. The exhaust steam in the design shown passes from the space *i* to the condenser; in some designs the condenser is placed directly in the base, that is, within the space *i*. A Curtis turbine having two wheels, as *d*, *d*, is called a two-stage turbine; if it has three wheels, it is spoken of as a three-stage turbine, and so on. Oil under pressure is forced by a pump into the step bearing, so that while running the shaft is supported on a film of oil. None of the oil used for lubrication mingles with the steam, which makes the condensed steam at once suitable for boiler feeding. All stationary blades do not extend around the circumference of the wheels *d*, *d*; the first row extends slightly beyond the nozzles; which are placed on one side of the machine; each succeeding row of stationary blades extends somewhat farther around the circumference and the final row or rows extend all around. As at present constructed,

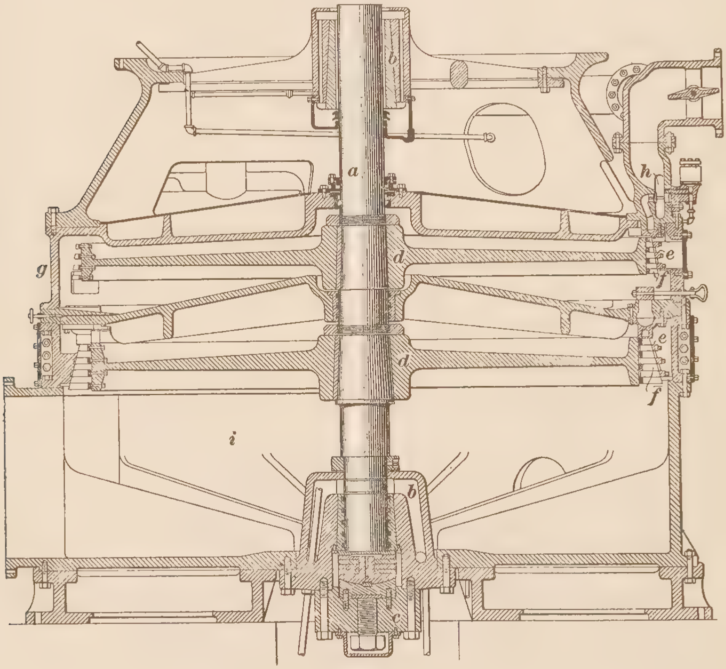

Fig. 10

the governor itself operates electromagnets, which in turn shut off successive nozzles, by closing their valves whenever the speed exceeds the desired limit. One of the nozzles has its controlling valve so arranged that the flow of steam through it can be throttled to any extent permitted by the construction; the other controlling valves are either opened or closed entirely by the governing mechanism.

GENERAL COMPARISONS

ADVANTAGES AND DISADVANTAGES

20. The fundamental difference in the operation of the common reciprocating-piston steam engine and of the steam turbine is that, while in the former the steam does work by reason of its pressure overcoming resistance, in the latter the steam does work by yielding its kinetic energy.

21. One of the greatest losses of heat in a reciprocating engine is due to the steam entering passages the walls of which have been cooled by contact with steam of much lower temperature. The weight of steam that is condensed and passed through a reciprocating engine without doing useful work, according to Dr. R. H. Thurston, amounts to about 25 per cent. of the total steam consumption of the engine. In the turbine this loss is entirely eliminated. The walls of the steam passages in the turbine soon attain a temperature equal to the temperature of the passing steam, and as the temperature of the steam at any given point does not vary, it follows that there is no transfer of heat between the steam and surrounding metal. This causes a practically ideal expansion of the steam.

22. A loss of energy to which the turbine is subject and that has no counterpart in the reciprocating engine may be caused by moisture in the steam; thus, a drop of water falls on the wheel and is given a velocity equal to that of the blades. As soon as it attains this velocity, centrifugal force causes it to fly off against the casing, where its kinetic energy is dissipated; falling back on the wheel it goes through the same cycle of operations again and continues to abstract energy from the revolving blades. Dr. R. H. Thurston, experimenting with a 10-inch De Laval turbine running at 20,000 revolutions per minute, estimates the loss from this source to be 1 horsepower for every 275 drops of water, each having a diameter of $\frac{1}{10}$ inch.

Another loss due to moisture in the steam is the retardation of the wheel caused by a thread-like stream of water between the revolving disk and the sides of the casing. The same author, experimenting with the same turbine, estimates the loss due to a thread of water $\frac{1}{1000}$ inch in width between the casing and the wheel to be $1\frac{1}{2}$ horsepower per hour.

23. The obvious remedy for loss due to moisture is the use of highly superheated steam, which is possible, since, unlike the reciprocating engine, the turbine has no rubbing surfaces on which lubrication is made difficult by steam of a high temperature. The trouble due to charred and burnt packing in stuffingboxes is also obviated, as there is no packing around the turbine to burn. With the reciprocating engine the limit to which superheating may be carried is set by the engine, while with the turbine the limit is set by the superheater. Experiments with a De Laval turbine conducted at Sibley College, Cornell University, show a gain of 1 per cent. in efficiency for every 30° of superheat. Other experiments with other turbines show a gain of 20 per cent. in economy with 60° superheat.

24. The efficiency of steam turbines measured by the water consumption per horsepower equals and in some cases excels that of the best types of modern reciprocating engines. The use of oil is much less than with the more common type of engine, as there are comparatively few bearings to oil, and no cylinder oil is required, owing to the absence of internal rubbing surfaces. This latter fact is quite a decided advantage when exhaust steam is condensed and returned to the boiler, since the water will be entirely free from oil.

Turbines are worked with a very high ratio of expansion, which is frequently as high as 1 : 150.

HORSEPOWER OF STEAM TURBINES

25. The horsepower developed by a steam turbine cannot be readily computed from an indicator diagram, as is done with reciprocating engines; in practice it is measured by means of some form of a dynamometer, as a Prony brake, for instance. For this reason, whenever the horsepower of a steam turbine is mentioned, the brake, or actual, horsepower is meant. This fact must be clearly borne in mind when making a comparison of the horsepowers of a reciprocating engine and a steam turbine, since in the reciprocating engine the indicated horsepower is always given, unless distinctly stated otherwise. The indicated horsepower of such an engine is always much more, probably by 20 per cent. on an average, than the actual delivered horsepower, the difference being required to overcome frictional resistances within the engine.

A SERIES

OF

QUESTIONS AND EXAMPLES

RELATING TO THE SUBJECTS
TREATED OF IN THIS VOLUME.

It will be noticed that the Examination Questions that follow have been divided into sections, which have been given the same numbers as the Instruction Papers to which they refer. No attempt should be made to answer any of the questions or to solve any of the examples until that portion of the text having the same section number as the section in which the questions or examples occur has been carefully studied.

PRINCIPLES OF MECHANICS.

EXAMINATION QUESTIONS.

(1) Define mechanics.

(2) Define (*a*) matter; (*b*) molecule; (*c*) atom.

(3) In what three states does matter exist?

(4) Explain clearly the distinction between general properties of matter and special properties of matter.

(5) Define (*a*) motion; (*b*) velocity; (*c*) uniform velocity; (*d*) variable velocity.

(6) Define (*a*) acceleration; (*b*) retardation; (*c*) average velocity.

(7) An ocean steamer made a run of 3,240 miles in 6 days and 16 hours. What was its average speed in miles per hour? Ans. $20\frac{1}{4}$ mi.

(8) How long would it take to make a tour around the world when traveling at an average speed of 3,000 feet per minute, assuming the distance around the world to be 25,000 miles? Ans. 30 da. 13 hr. 20 min.

(9) How do we recognize the existence of a force?

(10) What conditions are necessary to compare the relative effects of different forces on different bodies?

(11) State Newton's Laws of Motion.

(12) What do you understand by the term "inertia"?

(13) Define (*a*) dynamics; (*b*) statics.

(14) Why is not the weight of a given body the same at every point on the surface of the earth?

(15) Determine the mass of a body that weighs 346 pounds at a place where g is equal to 32.174. Ans. 10.75+.

(16) How does the position of a body above or below the surface of the earth affect its weight?

(17) A locomotive weighing 30 tons has to overcome a constant force of 15 pounds per ton when it is in motion. What total force must the locomotive exert so that its speed may increase at the rate of 3 feet per second?

Ans. 6,047 lb.

(18) What will be the momentum of the locomotive in the preceding example when it has attained a velocity of 1 mile per hour? Ans. 2,736.3 lb.

(19) Define (*a*) work; (*b*) power; (*c*) energy.

(20) What horsepower is required to raise a body weighing 66,000 pounds through a distance of 80 feet in $\frac{1}{2}$ hour?

Ans. $5\frac{1}{3}$ H. P.

(21) A body weighing 6,432 pounds is moving with a constant velocity of 60 feet per second. What horsepower will be required to bring the body to rest in 3 minutes?

Ans. $3\frac{7}{11}$ H. P.

(22) What is the tangential pressure on the crank of an engine when the crank is on either dead center?

(23) What is friction?

(24) A body weighing 5,000 pounds rests on a horizontal surface. In order to slide the body along the surface, a horizontal force of 300 pounds must be exerted. What is the coefficient of friction in this particular case? Ans. .06.

(25) A crosshead weighing 1,000 pounds and having bronze shoes slides on a slightly greased, horizontal wrought-iron guide. If the contact area of the bronze shoe is 100 square inches, what will be (*a*) the total friction and (*b*) the friction per square inch of contact surface?

$$\text{Ans.}\begin{cases}(a) & 160\text{ lb.}\\(b) & 1.6\text{ lb.}\end{cases}$$

(26) What is the center of gravity of a body?

(27) Explain the method of finding the common center of gravity of several bodies whose weights and the distances between whose centers of gravity are known.

(28) Give a practical method of determining the center of gravity of a solid body.

(29) Define (*a*) centrifugal force; (*b*) centripetal force.

(30) What is the relation between the centrifugal and centripetal forces of a revolving body?

(31) A body weighing 10 pounds revolves at a speed of 60 revolutions per minute about a point 6 feet from its center of gravity. What is the centrifugal force tending to pull the body from the point about which it revolves?

Ans. 73.44 lb.

(32) Name the three states of equilibrium and give examples of each.

(33) In the case of a body at rest, what are the conditions of the forces acting on that body?

(34) How is the condition of equilibrium of the forces acting on a body affected by the position of its "line of direction" with respect to the base?

MACHINE ELEMENTS.

EXAMINATION QUESTIONS.

(1) Define (*a*) lever; (*b*) weight arm; (*c*) force arm; (*d*) fulcrum.

(2) What is the condition necessary for the equilibrium of the lever?

(3) State the general rule that expresses the relation existing between the weight, the force, and the distances through which they move.

(4) What must be the length of the weight arm in order that a force of 12 pounds at a distance of 20 inches from the fulcrum will raise a weight of 100 pounds at the end of the weight arm? Ans. 2.4 in.

(5) Into what two classes may pulleys be divided in reference to their construction?

(6) What advantages have split pulleys over solid pulleys?

(7) Explain how a crowned pulley tends to prevent the slipping off of the belt.

(8) What is meant by "balancing pulleys," and how is it accomplished?

(9) Define the terms "driver" and "driven" as applied to pulleys.

(10) Find what diameter driver will be required which, when running at 600 revolutions per minute, will cause the driven, whose diameter is 6 inches, to run at 1,800 revolutions per minute. Ans. 18 in.

(11) What must be the speed of a driver 12 inches in diameter in order that the driven, whose diameter is 5 inches, may make 1,600 revolutions per minute?
Ans. $666\frac{2}{3}$ rev. per min.

(12) The distance between the centers of two pulleys, whose diameters are 6 feet and 2 feet, respectively, is 40 feet. What is the required length of an open belt? Ans. 93 ft.

(13) A single belt running at 1,650 feet per minute is used to transmit 40 horsepower. If the arc of contact on the small pulley is 120°, how wide should the belt be?
Ans. 28 in.

(14) What horsepower will be transmitted by the belt in question 13 if the velocity is reduced to 1,200 feet per minute? Ans. 29.3 H. P.

(15) If, in question 13, a double belt were substituted for the single belt, how wide should it be? Ans. 20 in.

(16) Which side of a leather belt should be in contact with the pulley face, and why?

(17) Should rosin be used on a belt to prevent slipping?

(18) Give the causes of flapping belts and their remedies.

(19) State the different methods used in joining belts. Which of these makes the best joint?

(20) What precaution should be observed when using rubber belts?

(21) Define the terms "driver" and "follower" as applied to a train of gear-wheels.

(22) In Fig. 19, if the diameter of A is 90 inches, that of F 30 inches, and if B has 12 teeth, C 30 teeth, D 20 teeth, and E 36 teeth, find the weight that a force of 50 pounds at P can raise. Ans. 675 lb.

(23) What is (*a*) circular pitch? (*b*) diametral pitch?

(24) What are the most common forms of teeth used in ordinary practice?

(25) What advantages have involute teeth over epicycloidal teeth?

(26) Find the pitch diameter of a gear-wheel having 60 teeth and a circular pitch of 1.152 inches. Ans. 22 in.

(27) What is the circular pitch of a gear-wheel 30 inches in diameter having 60 teeth? Ans. 1.57 in.

(28) What is the over-all diameter of a gear-wheel having 80 teeth with a diametral pitch of 8? Ans. 10¼ in.

(29) Find the number of teeth in a gear-wheel whose outside diameter is 7½ inches and whose diametral pitch is 8. Ans. 58 teeth.

(30) What distinguishes a fixed pulley from a movable pulley?

(31) In a set of pulleys there are fourteen parts of the rope supporting the load. Neglecting friction losses, what weight can a force of 150 pounds raise, when applied to the free end of the rope? Ans. 2,100 lb.

(32) How does friction affect the force required to raise a given weight by means of a rope or chain and pulleys?

(33) In a block and tackle having eight parts of the rope supporting the load, what probable actual force is required to raise 1,500 pounds? Ans. 312.5 lb.

(34) In what respect is the Weston differential pulley block better than the ordinary block and tackle?

(35) An inclined plane is 70 feet long and 12 feet high. What force acting parallel to the plane will be required to sustain a weight of 600 pounds on the plane? Ans. 103 lb.

(36) What weight will a force of 36 pounds acting parallel to the base of the plane be able to sustain on an inclined plane having a base 50 feet long and a height of 15 feet? Ans. 120 lb.

(37) What is the probable actual weight that can be raised by means of a screw jack that has a screw $2\frac{1}{2}$ inches in diameter with 6 threads to the inch if a force of 50 pounds is applied at the end of a lever 20 inches from the shaft?

Ans. 2,356.2 lb.

(38) Define (*a*) velocity ratio; (*b*) efficiency.

(39) What is the efficiency of the screw jack of question 37? Ans. $6\frac{1}{4}$ per cent.

MECHANICS OF FLUIDS.

EXAMINATION QUESTIONS.

(1) Define hydrostatics.

(2) How can it be proved that liquids transmit pressure in all directions and with the same intensity?

(3) State Pascal's law.

(4) In what direction does the pressure due to the weight of a body of water act in a vessel in which water is contained?

(5) What is the pressure on the bottom of a vessel if the base is a circle 3 inches in diameter, the height 8 inches, and the vessel is completely filled with water? Ans. 2.045 lb.

(6) State the law for the upward pressure of a liquid on a horizontal surface submerged in the liquid.

(7) Why is it that in several pipes that communicate with one another and that differ in shape and size, water will stand at the same height?

(8) Why is it that water issuing from a hose connected to a hydrant cannot be made to spout up to a level with the surface of the water in the reservoir that supplies the hydrant?

(9) Define specific gravity.

(10) Calculate the weight of a block of aluminum whose volume is 100 cubic feet. Ans. 15,605 lb.

(11) State the principle of Archimedes.

(12) Explain how, by the application of the principle of Archimedes, the volume of an irregularly shaped body may be accurately determined.

(13) (*a*) What are hydrometers? (*b*) Into what two classes may hydrometers be divided?

(14) A single-cylinder pump feeds a boiler through a delivery pipe 1 inch in actual diameter. The piston speed is such as to give a velocity of flow of 400 feet per minute. How many gallons of water can be pumped into the boiler in 1 hour? Ans. 979.2 gal.

(15) What should be the commercial size of a delivery pipe from a duplex pump to deliver 936 gallons of water per hour? Ans. 1 in.

(16) Why is it important to have as few bends as possible in the suction pipe leading to a pump?

(17) What simple experiment proves that gases tend to expand and increase their volume?

(18) How high a column of mercury will the pressure of the atmosphere support?

(19) How can the degree of the vacuum in a vessel be determined?

(20) How high a column of a liquid whose specific gravity is 2.5 will the atmospheric pressure support? Ans. 163.2 in.

(21) How is the pressure of the atmosphere measured?

(22) Why does the pressure of the atmosphere decrease as the elevation above sea level increases?

(23) In what respect is the action of the pressure of the atmosphere similar to that of the pressure of a liquid?

(24) How does the tension of a gas change with the change in volume under constant temperature?

(25) Define (*a*) gauge pressure; (*b*) absolute pressure.

(26) A metallic tube closed at one end is fitted with an air-tight movable piston. When the piston is at the open end of the tube, the pressure of the air within the tube is equal to 14.7 pounds per square inch. What will be the pressure of the air between the piston and the closed end of the tube after the former has moved toward the latter a distance equal to $\frac{4}{5}$ the length of the tube, the temperature remaining constant? Ans. 73.5 lb. per sq. in.

(27) Suppose that the piston in the tube mentioned in question 26 had been moved toward the closed end until the pressure had reached 147 pounds per square inch. What fraction of the original volume would the compressed air have occupied? Ans. $\frac{1}{10}$.

(28) In pneumatics, what is an air pump?

(29) Can a perfect vacuum be produced with the air pump?

(30) Explain the operation of the dashpot of a Corliss engine.

(31) Explain the principle of operating the siphon.

(32) Suppose that two vessels, in one of which the water stands at a higher level than in the other, are connected by a siphon and water is siphoned from the vessel in which it stands at the higher level into the other vessel. What will happen when the water in each vessel reaches the same level?

(33) In practice, how far above the water level in the vessel from which water is siphoned can the highest point of the siphon be carried?

(34) (*a*) What is a pump? (*b*) Into how many types may pumps be divided in reference to their mode of action? (*c*) Name these types.

(35) Explain the principle on which the suction pump acts.

(36) What advantage has a lifting pump over a suction pump?

(37) In what respect does a force pump differ from a lifting pump?

(38) What is the difference between a single-acting and a double-acting force pump?

STRENGTH OF MATERIALS.

EXAMINATION QUESTIONS.

(1) (*a*) Define stress. (*b*) Name the various kinds of stresses to which a body can be subjected.

(2) A weight of 8,000 pounds rests on the top of a cubical block of wood the area of each face of which is 20 square inches. What is the unit stress, in pounds per square inch, to which the block is subjected? Ans. 400 lb. per sq. in.

(3) Define (*a*) strain; (*b*) elasticity; (*c*) elastic limit.

(4) Taking the ultimate tensile strength of wrought iron as 55,000 pounds per square inch, what tensile force will rupture a wrought-iron bar whose cross-sectional area is 4 square inches? Ans. 220,000 lb.

(5) How does annealing improve old chains?

(6) Give a practical method of determining the true condition of a given rope, supposing that its outer surface appears to be in good condition.

(7) (*a*) On what does the safe lifting load of a sling depend? (*b*) Explain how the style of attachment to the hook of the tackle block affects the amount of load to be raised.

(8) (*a*) For what is manila rope chiefly used? (*b*) What is the object of lubricating the fibers of a rope?

(9) In general, how should the size of the pulley compare with the size of manila rope used for transmitting power?

(10) What is the greatest load to which an iron-wire rope 1½ inches in circumference should be subjected?
Ans. 1,350 lb.

(11) What should be the circumference of a steel-wire rope under a maximum working load of 16,000 pounds?
Ans. 4 in., nearly.

(12) How does the strength of a column having both ends flat compare with that of columns both of whose ends are not flat but which in every other respect are similar to the first column?

(13) Give practical examples of the three different classes into which columns are divided with respect to the condition of their ends.

(14) What is the safe steady working load that a cast-iron column, having fixed ends, 14 inches in diameter and 16 feet high can sustain? Ans. 1,291,480 lb.

(15) Would you consider a steel piston rod 6 inches in diameter of sufficient size for a 40-inch cylinder using steam at 110 pounds pressure? If so, why?

(16) A solid yellow-pine beam 14 inches square rests on two supports 12 feet apart. What steady safe load will the beam support at its middle point? Ans. 18,547 lb.

(17) An oak beam 4 inches wide and 6 inches deep is subjected to a sudden shearing stress of 10,000 pounds across the grain. Is this a safe load for the given conditions?

(18) What should be the area of a wrought-iron beam to safely support a sudden shearing stress of 400,000 pounds?
Ans. 91 sq. in., nearly.

(19) What do you understand by double shear?

(20) What is the use of countershafts?

(21) What is the distinction between cold-rolled shafting and bright shafting?

(22) How is bright shafting designated commercially with respect to size?

(23) Why is it good practice to place pulleys for transmitting or receiving power as near the bearings of the shaft as possible?

(24) What horsepower can be transmitted from a steel shaft 8 inches in diameter by means of pulleys between the bearings when running at a speed of 80 revolutions per minute? Ans. 482 H. P.

(25) What must be the speed of a 4-inch shaft of cold-rolled iron, having no pulleys between bearings, to transmit 75 horsepower? Ans. 76 rev.

(26) Find the diameter of a wrought-iron shaft running at 300 revolutions per minute to transmit 84 horsepower by means of pulleys between its bearings. Ans. 3 in.

(27) How does a change in the speed of a shaft affect the amount of power transmitted?

(28) Does a high tensile strength in metals necessarily imply an ability on the part of the metal to safely resist repeated applications of sudden stresses?

(29) What is the chief advantage of a steel rope over an iron rope?

HEAT AND STEAM.

EXAMINATION QUESTIONS.

(1) What are the effects of heat on a substance when applied to it?

(2) What do you understand by the temperature of a body?

(3) (*a*) Define the term British thermal unit. (*b*) Give the mechanical equivalent of one unit.

(4) Give an instance of work being changed to heat.

(5) Define (*a*) latent heat of fusion; (*b*) latent heat of steam; (*c*) specific heat.

(6) It requires 1.77 B. T. U. to raise the temperature of 10 pounds of glass from 74° to 75°. What is the specific heat of glass? Ans. .177.

(7) One thousand pounds of steam at 212° is conveyed by pipes through a building; all the steam is condensed and the water of condensation returns to the boiler at a temperature of 192°. If the heat given off is applied to a quantity of water, how many pounds will it raise from 32° to 112°? Ans. 12,325 lb.

(8) If we place 10 pounds of water at a temperature of 90°, 15 pounds of mercury at 60°, and 20 pounds of alcohol at 40°, together in a vessel, what will be the temperature of the mixture? Ans. 62.66°

NOTE.—The specific heat of alcohol is .60.

(9) If a number of substances of different temperatures are mixed together, how does the quantity of heat in the mixture compare with the quantity of heat contained in all the substances before mixing?

(10) If 20 pounds of steam at 212° are mixed with 400 pounds of water at 55°, what will be the temperature of the mixture? Ans. 108.47°

(11) How does change of pressure on the surface of a liquid affect its boiling point?

(12) Define (*a*) saturated steam; (*b*) superheated steam.

(13) Define (*a*) heat of the liquid; (*b*) latent heat; (*c*) total heat; (*d*) specific volume; (*e*) density.

(14) How many B. T. U. are required to change 10 pounds of ice at 25° into steam at 212°? Ans. 12,935.28 B. T. U.

(15) What is the difference between gauge pressures and absolute pressures, and which are given in the Steam Tables?

(16) How many foot-pounds of work will it require to change 40 pounds of water at 90 pounds gauge pressure into steam at the same pressure? Ans. 27,428,172 ft.-lb

THE STEAM ENGINE.

EXAMINATION QUESTIONS.

(1) Define the terms head end and crank end of a steam engine cylinder.

(2) What is meant by the stroke of an engine?

(3) What are the stationary parts of an engine?

(4) Define *valve gear*.

(5) Explain the difference, if there is any, between piston clearance and clearance volume.

(6) What is understood by the throw of an eccentric?

(7) Explain the difference between outside lap and inside lap of a **D** slide valve.

(8) Is it possible to cut off when a **D** slide valve operated by an eccentric has no outside lap and no lead?

(9) What is meant by angle of advance?

(10) Suppose you had a plain slide-valve engine and you wished to make the cut-off earlier, what would you do? The port opening is to be the same as before.

(11) What is the effect of increasing the inside lap of a **D** slide valve?

(12) Define *lead*.

(13) With an ordinary slide valve and an engine running under, is the eccentric set behind or ahead of the crank?

(14) How can the valve be given a travel greater than the throw of the eccentric?

(15) If a reversing rocker is used with an ordinary slide valve, will the eccentric occupy the same position as with a direct rocker ?

(16) How does the angularity of the connecting-rod affect compression ?

(17) What is the object of the passage cored in an Allen valve ?

(18) State in your own words how to set the valve of a plain slide-valve engine.

(19) A 14″ × 28″ engine has a clearance volume of 247 cubic inches. Express the clearance in per cent.

Ans. 5.73 per cent.

(20) In a 36″ × 60″ engine the steam is cut off when the piston has moved over 21 inches of its stroke. The clearance being 2 per cent., find the real cut-off.

Ans. 36.27 per cent.

(21) What is the ratio of expansion of the engine given in question 20 ? Ans. 2.76.

(22) Suppose that the outside lap of a **D** slide valve is decreased, but that the valve travel and angle of advance remain the same as before. Investigate the effect of this with the aid of the Bilgram valve diagram and state your conclusions and explain how they were reached.

(23) A 44″ × 80″ engine is to run at 75 revolutions per minute. What actual diameter of steam and exhaust pipe should be used ?

$$\text{Ans.}\begin{cases}\text{Steam pipe} = 18 \text{ in.}\\ \text{Exhaust pipe} = 22 \text{ in.}\end{cases}$$

(24) What should be the area of the steam port for the engine given in question 23 if the steam port is short ?

Ans. 202.7 sq. in.

THE INDICATOR

EXAMINATION QUESTIONS.

(1) What is meant by the scale of an indicator spring?

(2) About what scale of spring is usually selected for a given boiler pressure?

(3) What is a reducing motion?

(4) What is the principal objection to the lazy-tongs and pantograph reducing motions?

(5) In Fig. 9, find the length of the arm UV so that the diagram may be 3 inches long, the stroke of the engine being 24 inches, and the length of the arm UW 40 inches.

Ans. 5 in.

(6) What precautions should be taken before attaching an indicator to an engine?

(7) What is the vacuum line and how is its position located?

(8) What are the distinguishing characteristics of indicator diagrams taken from Corliss engines as compared with diagrams taken from high-speed engines?

(9) (*a*) In a plain slide-valve engine, how would you remedy too early admission? (*b*) What effect would this remedy have on the other events of the stroke?

(10) If one end of a cylinder with a slide valve is found to be doing more work than the other, how can the fault be remedied?

(11) (*a*) How is the amount of compression influenced by the speed of the engine? (*b*) What should be the amount of compression for high-, low-, and medium-speed engines?

(12) What may be inferred (*a*) when the steam line falls abruptly? (*b*) when the back-pressure line is much above the atmospheric line? (*c*) when the actual expansion line rises above the theoretical expansion line?

(13) (*a*) What faults in steam distribution are shown by the diagram, Fig. I, which is taken from a plain slide-valve engine? (*b*) How may they be partially remedied?

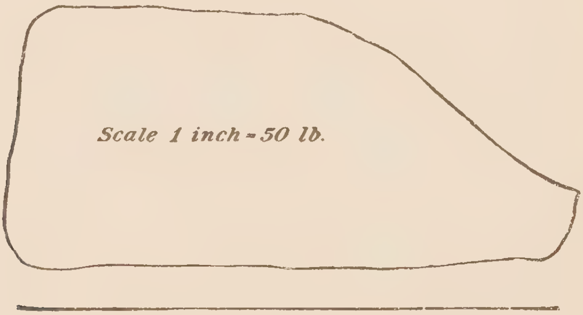

FIG. I.

(14) Criticize the indicator diagrams shown in Fig. II.

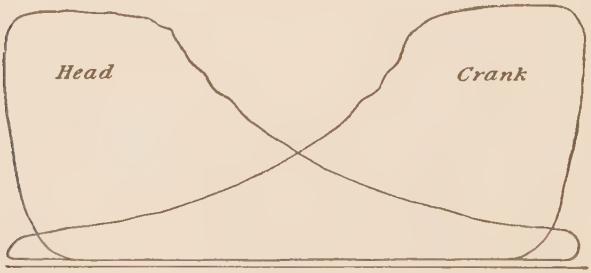

FIG. II.

(15) Why does the actual expansion line generally rise above the theoretical expansion line near the end of the stroke?

(16) What is the most general method of determining the point of cut-off on a diagram taken from a high-speed engine?

(17) (*a*) To what are wavy lines on a diagram generally due? (*b*) Expansion lines that drop by a series of steps?

(18) Criticize the diagrams shown in Fig. III.

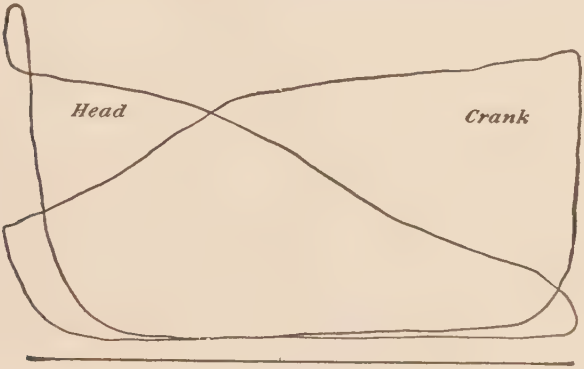

Fig. III.

(19) What is the cause of the difference in the shape of the loop above the steam line in Figs. 20 and 22?

(20) If the actual expansion line follows the theoretical expansion line closely, is that a positive indication that the valves and piston do not leak?

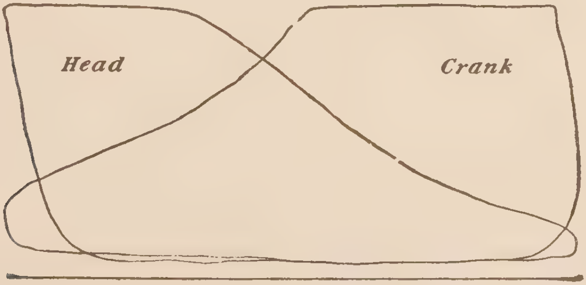

Fig. IV.

(21) Criticize the diagrams shown in Fig. IV.

ENGINE TESTING.

EXAMINATION QUESTIONS.

(1) Define (*a*) adiabatic expansion; (*b*) isothermal expansion.

(2) What relation exists between the amount of work done in compressing a gas isothermally or adiabatically and the amount of work done by the gas when expanding under similar conditions?

(3) Why is not the relation of volume and pressure of steam when it expands as simple as in the case of a perfect gas?

(4) If the net pressure on the piston of an engine is 45.6 pounds per square inch and the volume swept through by the piston at each stroke is 6.3 cubic feet, (*a*) how much work is done at each stroke? (*b*) The engine makes 76 strokes per minute; what horsepower does it develop?

$$\text{Ans.}\begin{cases}41{,}368.32\text{ ft.-lb.}\\ 95.27\text{ H. P.}\end{cases}$$

(5) What is (*a*) the mean ordinate of an indicator diagram and (*b*) how is it found?

(6) A diagram like that shown in Fig. 7 is 3 inches long and has an area of 6.75 square inches; the vertical scale of pressure is 50 pounds per inch; the cylinder from which the diagram was made has an area of 1 square foot and a length of 2 feet. (*a*) Find the horizontal scale of volumes and (*b*) the work per stroke of piston.

$$\text{Ans.}\begin{cases}(a)\quad \tfrac{2}{3}\text{ cu. ft. per in.}\\ (b)\quad 32{,}400\text{ ft.-lb.}\end{cases}$$

(7) How can the net horsepower of an engine be approximately obtained without the use of a dynamometer?

(8) The I. H. P. of an engine running under full load is 176.8. When running light the I. H. P. is 25.6. What is the efficiency of the engine? Ans. 85.5 per cent.

(9) The area of a diagram is 2.75 square inches and the length is 3.15 inches. A 50-pound spring was used. Find the M. E. P. Ans. 43.65 lb. per sq. in.

(10) In finding the area of a diagram with a planimeter, how may the accuracy of the work be easily checked?

(11) If the M. E. P. of a diagram with loops is to be found by the use of ordinates, how can the mean ordinate be found?

(12) Find the approximate M. E. P. of a non-condensing engine cutting off at $\frac{5}{8}$ stroke and making 300 revolutions per minute. The boiler pressure is 75 pounds gauge.

Ans. 59.54 lb. per sq. in.

(13) (*a*) What is meant by piston speed? (*b*) An engine with a 24-inch stroke runs at a speed of 180 revolutions per minute. What is the piston speed? Ans. 720 ft. per min.

(14) What is an engine constant?

(15) (*a*) What is the engine constant for a uniform speed of rotation of an 18″ × 24″ engine running at a speed of 185 R. P. M.? (*b*) What is the I. H. P. of the engine when the average M. E. P. for a pair of indicator diagrams is 53.8 pounds per square inch?

Ans. { (*a*) 5.706.
(*b*) 306.98.

(16) (*a*) Find the engine constant for a uniform scale, number of ordinates and piston speed of a 24″ × 36″ engine running at 150 revolutions per minute when 20 ordinates are used and the scale of the spring is 60. (*b*) What is the I. H. P. of the engine when the sum of the 20 ordinates is 16 inches?

Ans. { (*a*) 37.013.
(*b*) 592.2.

(17) What is meant by the brake horsepower of an engine?

(18) In Fig. 17 the distance from the center of the shaft to the point of support of the brake arm on the scale is 4 feet. When the brake is not in operation the scale balances at 14.5 pounds. What horsepower is developed by the engine when it is running at 225 revolutions per minute and the scale balances at 274.5 pounds? Ans. 44.55 H. P.

(19) In Fig. 18 the diameter of the pulley is 47 inches and the diameter of the rope is 1 inch. When the engine is running at 350 revolutions per minute, the weight W is 241 pounds and the spring balance A indicates 10 pounds. What horsepower is developed? Ans. 30.78 H. P.

(20) Why is the steam consumption, as calculated from an indicator diagram, always less than the actual steam consumption?

(21) How can an idea of the amount of cylinder condensation be obtained?

(22) The following measurements were taken from a diagram like Fig. 19: $a m = .70$ inch, $O m = 3.41$ inches, $b n = .62$ inch, $O n = .36$ inch, and $e h = 3.25$ inches. The diagram was taken from an engine having a 16″ × 20″ cylinder and running at 160 revolutions per minute. The area of the diagram is 2.41 square inches and the scale of the spring is 45 pounds. Find the steam consumption per I. H. P. per hour. Ans. 30.34 lb.

(23) The following measurements were obtained from a diagram (see Fig. 20) taken from a 24″ × 36″ engine: $a m = .71$ inch, $l = 2.93$ inches, $L = 3.36$ inches, M. E. P. = 37.5 pounds, spring 50. What is the steam consumption per I. H. P. per hour? Ans. 27.79 lb.

GOVERNORS.

EXAMINATION QUESTIONS.

(1) Explain how the flywheel aids the governor in taking care of sudden changes in the load.

(2) What two methods are generally used to vary the average work done in the cylinder?

(3) In Fig. 1, when will the ball be in equilibrium?

(4) Why is it inadvisable to use the simple pendulum governor on high-speed engines?

(5) What is meant by increasing the sensitiveness of a governor?

(6) (*a*) Why does a spring-loaded governor act more quickly than a weighted-pendulum governor, and (*b*) why is it less apt to hunt or race?

(7) If the eccentric is shifted at right angles to the center line of the crank, what is the maximum port opening when the valve has its least travel?

(8) (*a*) How is the lead affected by swinging the eccentric from a pivot on the center line of the crank and on the same side of the shaft as the crankpin? (*b*) Mention an advantage and a disadvantage of this method of suspension.

(9) How can the eccentric be pivoted so as to keep the lead nearly constant through the ordinary range of motion?

(10) In an inertia governor like that used on the McEwen engine, how does the inertia of the weights assist in adjusting the governor to changes in the speed?

(11) How can the inertia governor be used if it is desired to suspend the eccentric as in Fig. 13?

(12) Why will a perfectly isochronous governor rapidly fluctuate from one extreme position to another?

(13) How can the sensitiveness of an isochronous governor, of the revolving-pendulum or shaft-governor type, be reduced so as to be of practical use?

(14) Why does friction tend to cause a governor to hunt?

(15) An engine is running at 162 revolutions per minute, the governor pulley being 5 inches in diameter; what must be the diameter of the pulley if the engine is to run at 180 revolutions per minute? The governor is a simple pendulum governor. Ans. $5\frac{9}{16}$ in., nearly.

(16) Why is it desirable to run a spring-loaded governor as nearly as possible at the speed for which it was designed, instead of increasing the tension on the springs, to change the speed of the engine?

(17) Can the height of a simple pendulum governor, running at a given speed, be changed by changing the mass of the weights?

(18) How must the mass of the weights on a shaft governor be changed to increase the speed of the engine?

(19) What do you understand by a self-locking governor?

(20) If the eccentric is shifted at right angles to the center line of the crank, what effect does it have on the travel of the valve and the angle of advance of the eccentric?

(21) What condition must be fulfilled in order to make a governor perfectly isochronous?

(22) How can the springs of an inertia governor be proportioned so as to cause the engine to run faster under load than when running light?

VALVE GEARS.

(PART 1.)

EXAMINATION QUESTIONS.

(1) What are the leading types of valves?

(2) What is the object of using expansion valves?

(3) Mention two disadvantages of the Gonzenbach expansion valve.

(4) In what position relative to the crank are the eccentrics set for the Meyer cut-off valve?

(5) If the auxiliary valve of a riding cut-off valve is non-adjustable, how may its eccentric be set?

(6) Mention some of the disadvantages of a plain **D** slide valve when used as an automatic cut-off engine.

(7) What advantages has the pressure-plate valve over the piston valve?

(8) What precaution is necessary in starting an engine with piston valves or pressure-plate valves?

(9) If live steam should leak past the packing strips of an Allen balanced valve, would the balancing of the valve be affected?

(10) What is the function of the dashpot on a Corliss engine?

(11) The motion imparted to the valves of a Corliss valve gear by a given angular motion of the wristplate varies through wide limits. How is this variation made to improve the action of the valves?

(12) If the load on a Corliss engine becomes so heavy that the admission valve is not released how will the valve work?

(13) Explain the necessity of lead and lap on Corliss valves.

(14) Why are Corliss valve gears sometimes made with two eccentrics?

(15) If two eccentrics are used on a Corliss valve gear, what is the effect on the opening of the steam valves if the steam eccentric is set too near the crank position?

(16) Mention some advantages of the high-speed Corliss valve gear as compared with the ordinary Corliss gear.

(17) Suppose the wristplate of a Corliss gear overtravels at one end of its motion; how can this fault be remedied? The rocker-arm is supposed to be properly set.

(18) What effect on the events of stroke has increasing the distance between the valve plates of a Meyer cut-off valve?

(19) What do you understand by "overtravel" of a valve?

(20) Why are multiple-ported valves used?

(21) On an engine using the Meyer cut-off valve, admission occurs too early on the crank end, but is correct on the head end. What must be done in order that admission may occur at the correct time at both ends of the cylinder?

STEAM TURBINES

EXAMINATION QUESTIONS

(1) Does the De Laval steam turbine use the steam expansively?

(2) What is the object of the flexible bearing fitted to the shaft of the De Laval turbine?

(3) Why are the gears of a De Laval turbine made with right-handed and left-handed helical teeth?

(4) What means have been adopted in condensing De Laval turbines to give a close regulation during a very large and sudden decrease of load?

(5) What is the object of the flexible shaft fitted to the De Laval turbine?

(6) What is accomplished by the De Laval expansion nozzle?

(7) Is there any difference in the manner in which expansion of the steam is obtained in the De Laval and the Westinghouse-Parsons steam turbines? If so, explain it.

(8) How is vibration at high speeds prevented in the Westinghouse-Parsons turbine?

(9) Is steam admitted in a continuous stream to a Westinghouse-Parsons turbine? If not, how is it admitted?

(10) How is the steam expanded in the Curtis turbine?

(11) Do the stationary blades of the Curtis turbine extend around the circumference of the wheels carrying the moving blades ?

(12) How does a steam turbine differ in the use of steam from a reciprocating engine ?

A KEY

TO ALL THE

QUESTIONS AND EXAMPLES

CONTAINED IN THE

EXAMINATION QUESTIONS

INCLUDED IN THIS VOLUME.

The Keys that follow have been divided into sections corresponding to the Examination Questions to which they refer, and have been given corresponding section numbers. The answers and solutions have been numbered to correspond with the questions. When the answer to a question involves a repetition of statements given in the Instruction Paper, the reader has been referred to a numbered article, the reading of which will enable him to answer the question himself.

To be of the greatest benefit, the Keys should be used sparingly. They should be used much in the same manner as a pupil would go to a teacher for instruction with regard to answering some example he was unable to solve. If used in this manner, the Keys will be of great help and assistance to the student, and will be a source of encouragement to him in studying the various papers composing the course.

PRINCIPLES OF MECHANICS.

(1) Broadly speaking, mechanics is the science that treats of forces and their effects on material bodies. See Art. **1.**

(2) See Arts. **2, 3,** and **4.**

(3) Solid, liquid, and gaseous. See Arts. **6, 7,** and **8.**

(4) General properties are such as are common to all substances; special properties are such as are possessed by certain substances only. See Arts. **12** and **24.**

(5) (*a*) Motion is that condition of a body that causes it to change its position in relation to some other body. See Art. **30.**

(*b*) Velocity is the rate of motion, that is, the distance passed through in a unit of time. See Art. **32.**

(*c*) Uniform velocity means passing over equal distances in equal intervals of time. See Art. **32.**

(*d*) Variable velocity means passing over equal distances in unequal intervals of time, or passing over unequal distances in equal intervals of time. See Art. **33.**

(6) (*a*) Acceleration is the rate of change of velocity. See Arts. **35** and **57.**

(*b*) Retardation denotes the rate of decrease of velocity See Arts. **36** and **58.**

(*c*) The average velocity is that uniform velocity that carries the body over the same distance in the same time as the variable velocity. See Art. **37.**

(7) Since 1 day contains 24 hours, the total number of hours consumed in the trip was $6 \times 24 + 16 = 160$ hours. The average speed per hour was, therefore, $3{,}240 \div 160 = 20\frac{1}{4}$ mi. Ans. See Art. **38.**

(8) Reducing the 25,000 miles to feet, we have $25{,}000 \times 5{,}280 = 132{,}000{,}000$ feet. The time in minutes would, therefore, be $132{,}000{,}000 \div 3{,}000 = 44{,}000$ minutes. Reducing the time to days, hours, and minutes, we get 30 da. 13 hr. 20 min. Ans. See Art. **40.**

(9) By the effects it produces on matter. See Art. **41.**

(10) The point of application, the direction of action, and the magnitude of each force must be known. See Art. **44.**

(11) See Art. **46.**

(12) Inertia is that property of a body by virtue of which the body always tends to remain in the particular state of rest or motion that it has at the moment considered. See Art. **48.**

(13) See Arts. **52** and **53.**

(14) The weight of a body is proportional to the force of gravity; and since the force of gravity is different for different localities, the weight of any body will differ likewise. See Art. **56.**

(15) By rule 4, Art. **60,**

$$m = \frac{W}{g} = \frac{346}{32.174} = 10.75+. \quad \text{Ans.}$$

(16) See Art. **62.**

(17) The constant opposing force is $30 \times 15 = 450$ pounds. 30 tons $= 30 \times 2{,}000 = 60{,}000$ pounds. The force f to produce the required acceleration will, by rule 5, Art. **63,** be $\frac{W}{g}a = \frac{60{,}000}{32.16} \times 3 = 5{,}597+$ pounds. Hence, the total force required will be $5{,}597 + 450 = 6{,}047$ lb. Ans.

(18) 1 mile per hour = 5,280 feet per hour. $\frac{5,280}{60 \times 60} = \frac{22}{15} = 1\frac{7}{15}$ feet per second. Momentum, according to Art. **67,** $= \frac{60,000}{32.16} \times \frac{22}{15} = 2,736.3$ lb. Ans.

(19) (*a*) Work is the overcoming of a resistance through a certain distance. See Arts. **69, 70,** and **71.**

(*b*) Power is the rate of doing work. See Arts. **72** and **73.**

(*c*) Energy denotes ability to do work. See Arts. **74** and **75.**

(20) $\frac{66,000 \times 80}{33,000 \times 30} = 5\frac{1}{3}$ H. P. Ans. See Art. **73.**

(21) The kinetic energy of the body by rule 7, Art. **76,** is $\frac{w v^2}{2 g} = \frac{6,432 \times 60^2}{2 \times 32.16} = 360,000$ foot-pounds. This amount of work must be done in 3 minutes. Hence, the horsepower required to stop the body will be $\frac{360,000}{3} \div 33,000 = 3\frac{7}{11}$ H. P. Ans.

(22) Zero. See Art. **86.**

(23) Friction is the resistance that a moving body encounters from the surface of another body along which or through which it moves. See Art. **88.**

(24) According to Art. **89,** the coefficient of friction will be $\frac{300}{5000} = .06$. Ans.

(25) (*a*) From Table I we find the coefficient of friction to be .16. Hence, the total friction will be $1,000 \times .16 = 160$ lb. Ans. See Art. **92.**

(*b*) $\frac{160}{100} = 1.6$ lb. per sq. in. Ans.

(26) The center of gravity of a body is a point at which the whole weight of the body may be considered as concentrated. This point may be either inside or outside the body. See Art. **98.**

(27) See Art. **99.**

(28) Suspend the body successively from two different points and find the point of intersection of two plumb-lines from the points of suspension. The center of gravity will be on the line passing through the point of intersection and perpendicular to the plumb-lines. See Art. **103.**

(29) (*a*) Centrifugal force is that force that tends to pull a revolving body away from the point about which the body revolves. See Art. **104.**

(*b*) Centripetal force is that force that tends to pull a revolving body toward the point about which it revolves. See Art. **105.**

(30) The centrifugal and centripetal forces of a given revolving body are always equal and opposite. See Art. **104.**

(31) Applying rule 14, given in Art. **106,**

$$F = .00034 \times 10 \times 6 \times 60^2 = 73.44 \text{ lb.} \quad \text{Ans.}$$

(32) See Arts. **110, 111,** and **112.**

(33) The forces acting on a body at rest are in equilibrium. See Art. **109.**

(34) As long as the line of direction falls within the base, the body will stand; if it falls outside the base, the body will fall. In the former case the forces will be in equilibrium; in the latter they will not be in equilibrium. See Art. **113.**

MACHINE ELEMENTS.

(1) (*a*) A lever is a rigid bar or rod capable of being turned about a pivot.

(*b*) The weight arm is that part of the lever between the fulcrum and the weight.

(*c*) The force arm is that part of the lever between the fulcrum and the force.

(*d*) The fulcrum is the point or pivot about which the lever turns. See Arts. **1** and **2.**

(2) The product of the weight and the weight arm must be equal to the product of the force and the force arm. See Art. **2.**

(3) See rule **1,** Art. **3.**

(4) $\frac{12 \times 20}{100} = 2.4$ in. Ans. See Art. **3.**

(5) Solid pulleys and split pulleys. See Art. **9.**

(6) Split pulleys can be more easily put on and removed from shafts than solid pulleys. See Art. **9.**

(7) As explained in Art. **11,** the belt climbs toward the highest part of the pulley; and since in a crowned pulley the highest portion is in the center, the belt will tend to stay on the pulley.

(8) "Balancing pulleys" is the operation of making opposite sides of pulleys equal in weight, so that the centrifugal

forces of the opposite sides of a revolving pulley will be equal. A pulley is balanced by first finding which side is the lighter and then adding weights to the lighter side until it is as heavy as the other. See Art. **12.**

(9) The "driver" is the pulley that imparts motion to the belt. The "driven" is the pulley that receives motion from the belt. See Art. **14.**

(10) Applying rule **3**, Art. **15,**

$$D = \frac{6 \times 1,800}{600} = 18 \text{ in.} \quad \text{Ans.}$$

(11) Applying rule **6**, Art. **18,**

$$N = \frac{5 \times 1,600}{12} = 666\tfrac{2}{3} \text{ rev. per min.} \quad \text{Ans.}$$

(12) Applying rule **7**, Art. **23,**

$$B = 3\tfrac{1}{4}\left(\frac{6+2}{2}\right) + 2 \times 40 = 93 \text{ ft.} \quad \text{Ans.}$$

(13) From Table I we find that the allowable effective pull C for an arc of contact of 120° is 28.8 pounds. Applying rule **8**, Art. **27,**

$$W = \frac{33,000 \times 40}{1,650 \times 28.8} = 27.7, \text{ say } 28 \text{ in.} \quad \text{Ans.}$$

(14) Applying rule **9**, Art. **28,**

$$H = \frac{28.8 \times 28 \times 1,200}{33,000} = 29.3+ \text{ H. P.} \quad \text{Ans.}$$

(15) Applying rule **10**, Art. **31,**

$$W = \frac{23,100 \times 40}{1,650 \times 28.8} = 19.44, \text{ say } 20 \text{ in.} \quad \text{Ans.}$$

(16) The hair or grain side is commonly considered to be the proper side to be in contact with the pulley face. For reasons see Art. **32.**

(17) No; rosin makes the belt gummy and causes it to crack. See Art. **32.**

(18) Pulleys run out of true; they should be turned true. Pulleys out of line; they should be lined up. Belts sometimes flap if they are run at speeds over 4,000 feet per minute; perforating the belt with a series of small holes is said to remedy the defect in this case. Lack of steadiness in running; take steps to insure steady running. A defective joint; unlace the joint and relace it properly. Too great a distance between pulleys; reduce the distance if possible or substitute a wider belt. See Art. **33.**

(19) Lacing, sewing, riveting, and cementing. The cemented joint is considered the best.

(20) See Art. **38.**

(21) A wheel that imparts motion to another is called a "driver." A "follower" is a wheel that receives motion from another.

(22) Letting W represent the weight we have, by rule **12**, Art. **39,**

$$50 \times 90 \times 30 \times 36 = W \times 30 \times 12 \times 20,$$

or

$$4{,}860{,}000 = 7{,}200\ W.$$

Hence,
$$W = \frac{4{,}860{,}000}{7{,}200} = 675 \text{ lb.} \quad \text{Ans.}$$

(23) (*a*) Circular pitch is the distance measured along the pitch circle from a point on one tooth to the corresponding point on the next tooth. See Art. **47.**

(*b*) Diametral pitch is the number of teeth per inch of pitch diameter. See Art. **47.**

(24) The epicycloidal and the involute. See Arts. **51** and **52.**

(25) Involute teeth are stronger and their action is more satisfactory. See Art. **52.**

(26) Applying rule **13**, Art. **54,**

$$D = \frac{1.152 \times 60}{3.1416} = 22 \text{ in.} \quad \text{Ans.}$$

(27) Applying rule **15,** Art. **56,**

$$P = \frac{3.1416 \times 30}{60} = 1.57 \text{ in.} \quad \text{Ans.}$$

(28) Applying rule **17,** Art. **59,**

$$D_o = \frac{80 + 2}{8} = 10\tfrac{1}{4} \text{ in.} \quad \text{Ans.}$$

(29) Applying rule **21,** Art. **63,**

$$N = 7.5 \times 8 - 2 = 58 \text{ teeth.} \quad \text{Ans.}$$

(30) A fixed pulley has a stationary block. A movable pulley has a movable block. See Arts. **71** and **72.**

(31) Calling W the weight that can be raised, we have, by applying rule **27,** Art. **74,**

$$W = 150 \times 14 = 2{,}100 \text{ lb.} \quad \text{Ans.}$$

(32) The force that must be applied is greater than if there were no frictional losses. See Art. **75.**

(33) Applying rule **28,** Art. **75,**

$$f_a = \frac{1{,}500}{8 \times .60} = 312.5 \text{ lb.} \quad \text{Ans.}$$

(34) With the Weston differential pulley block a much greater weight can be raised with a given force than with the ordinary pulley; and the load can be stopped anywhere by ceasing to pull on the chain. See Art. **78.**

(35) Applying rule **30,** Art. **82,**

$$P = \frac{600 \times 12}{70} = 102.8+, \text{ say } 103 \text{ lb.} \quad \text{Ans.}$$

(36) Applying rule **31,** Art. **83,**

$$W = \frac{36 \times 50}{15} = 120 \text{ lb.} \quad \text{Ans.}$$

(37) First applying rule **34,** Art. **89,** we find the factor by which the theoretical weight is to be multiplied.

Thus, $$E = \frac{\frac{1}{6}}{\frac{1}{6} + 2\frac{1}{2}} = \tfrac{1}{16}.$$

Then applying rule **35,** Art. **90,** to obtain the theoretical weight, we find

$$W = \frac{6.2832 \times 50 \times 20}{\frac{1}{6}} = 37{,}699.2 \text{ pounds.}$$

Finally applying the principle of Art. **92,** we find the probable actual weight to be

$$37{,}699.2 \times \tfrac{1}{16} = 2{,}356.2 \text{ lb.} \quad \text{Ans.}$$

(38) (*a*) The velocity ratio is the ratio between the distance through which the force acts and that through which the weight moves. See Art. **103.**

(*b*) The efficiency is the ratio of the actual work to the theoretical work. See Art. **105.**

(39) By Art. **105,** the efficiency would be $\frac{2{,}356.2}{37{,}699.2}$ $= .0625$, or $6\frac{1}{4}$ per cent. Ans.

MECHANICS OF FLUIDS.

(1) Hydrostatics is that branch of mechanics that treats of liquids at rest. See Art. **1.**

(2) See Art. **4.**

(3) See Art. **5.**

(4) The pressure due to the weight of water acts, like any other pressure to which the water may be subjected, in *all* directions. See Art. **5.**

(5) First find the area of the base. This equals $.7854 \times 3^2 = 7.0686$ square inches. Multiplying this by the height of the water, we get $7.0686 \times 8 = 56.5488$ cubic inches as the volume of a column of water whose weight is equal to the pressure on the base. Therefore, the pressure on the base $= 56.5488 \times .03617 = 2.045+$ lb. Ans. See Arts. **7, 8,** and **9.**

(6) See Art. **12.**

(7) The water stands at the same level in each tube, because it is then in equilibrium. See Art. **16.**

(8) Because of the resistance of the air and the friction in the pipe. See Art. **17.**

(9) Specific gravity is the ratio between weights of equal volumes of any substance and water. Thus, the specific gravity of iron is the ratio between the weight of a given volume of iron and the weight of an equal volume of water. See Art. **23.**

(10) Using the principle of Art. **25** and taking the specific gravity from Table I, we have for the weight of the aluminum $62.42 \times 2.50 \times 100 = 15{,}605$ lb. Ans.

(11) A body immersed in a liquid will be buoyed up with a force equal to the weight of the liquid displaced. See Art. **26.**

(12) See Art. **29.**

(13) (*a*) Hydrometers are instruments for determining the specific gravity of liquids and of some forms of solids.

(*b*) Hydrometers of constant weight and hydrometers of constant volume. See Art. **30.**

(14) Applying rule **5,** we have

$$n = \frac{.7854 \times 1^2 \times 400}{19.25} = 16.32 \text{ gallons.}$$

As there are 60 minutes in 1 hour, the amount of water that can be pumped in 1 hour will be $16.32 \times 60 = 979.2$ gal. Ans.

(15) The amount of water to be delivered per minute will be $\frac{936}{60} = 15.6$ gallons. The velocity of flow for this case should not exceed 500 feet per minute. Applying rule **4,** we have

$$A = \frac{19.25 \times 15.6}{500} = .601 \text{ square inch, nearly.}$$

From Table VI we see that the nearest larger size pipe is the 1-inch pipe, which has an actual area of .863 square inch. Hence, a 1-inch pipe should be used.

(16) Bends in a pipe greatly increase the resistance to the flow of water, and, hence, also the power required to move the water through the pipe. As the power available for the suction pipe of a pump is quite small, any increase in the resistance to the flow will materially interfere with the satisfactory working of the pump. See Art. **39.**

(17) See Art. **41.**

(18) About 30 in., as explained in Art. **43.**

(19) By means of a vacuum gauge. See Arts. **45** to **47**, inclusive.

(20) Applying the principle of Art. **48**, we have $\frac{30 \times 13.6}{2.5} = 163.2$ in. Ans.

(21) The pressure of the atmosphere is measured with an instrument called a barometer. See Arts. **49** to **52**, inclusive.

(22) The higher the elevation the shorter is the column of air and the lower the density of the air. Consequently, the pressure exerted by the air becomes less the higher we go above sea level. See Art. **53.**

(23) Like the pressure exerted by a liquid, the pressure of the atmosphere acts in all directions with the same intensity. See Art. **54.**

(24) The pressure of a gas varies inversely as the volume, the temperature remaining constant. Thus, if the volume is reduced one-half, the pressure is doubled; if the volume is reduced to one-third the original volume, the pressure is increased to three times the original pressure. See Art. **55.**

(25) (*a*) Gauge pressure is pressure measured above the atmospheric pressure. See Art. **56.**

(*b*) Absolute pressure is pressure measured from vacuum and is equal to gauge pressure plus atmospheric pressure. See Art. **56.**

(26) Call the original volume, or volume of the tube, 1; then the volume after the piston has moved $\frac{4}{5}$ the length of the tube will be $1 - \frac{4}{5} = \frac{1}{5}$. Applying rule **7**, we have

$$p_1 = \frac{14.7 \times 1}{\frac{1}{5}} = 73.5 \text{ lb. per sq. in.} \quad \text{Ans}$$

(27) Applying rule **8**,

$$v_1 = \frac{14.7 \times 1}{147} = .1 = \tfrac{1}{10} \text{ the original volume.} \quad \text{Ans.}$$

(28) An air pump is an apparatus for removing air from any vessel; in other words, it is an apparatus for producing a vacuum in any vessel. See Art. **60.**

(29) No. See Art. **61.**

(30) See Art. **62.**

(31) See Arts. **63** and **64.**

(32) When the water reaches the same level in both vessels, the flow of water from one vessel into the other will cease. This follows from Arts. **63** and **64.**

(33) The highest point of the siphon should not be more than 28 feet above the level of the water that is being siphoned, or the siphon will not work. See Art. **65.**

(34) (*a*) A pump is an apparatus for raising water. See Art. **68.**

(*b*) Pumps may be divided into three classes, depending on their mode of operation. See Art. **69.**

(*c*) (1) Suction pumps; (2) lifting pumps; and (3) force pumps. See Art. **69.**

(35) A suction pump acts by producing a vacuum in the suction pipe, which causes the water into which the lower end of the suction pipe is immersed to rise in the suction pipe. See Art. **70.**

(36) A lifting pump will raise water to a higher point than a suction pump. See Art. **72.**

(37) A force pump piston does not have any valve like the piston of a lifting pump. See Art. **73.**

(38) A single-acting force pump discharges water at every second stroke; a double-acting force pump discharges water during each stroke. See Art. **76.**

STRENGTH OF MATERIALS.

(1) (*a*) A stress is a force that acts in the interior of a body and tends to produce a deformation in the body. See Art. **1.**

(*b*) Tensile, compressive, shearing, transverse, and torsional. See Art. **2.**

(2) By Art. **3,** we have

$$\text{unit stress} = \frac{8000}{20} = 400 \text{ lb. per sq. in.} \quad \text{Ans.}$$

(3) (*a*) Strain is the amount of deformation produced in a body by the action of a stress. See Art. **4.**

(*b*) Elasticity is that property of a body by virtue of which it will tend to return to its original shape after the force producing the deformation is removed. See Art. **5.**

(*c*) The elastic limit is that unit stress at which permanent set begins. See Art. **7.**

(4) Since the ultimate tensile strength per square inch is 55,000 pounds, the force at which a bar having an area of 4 square inches will rupture is 4 × 55,000 = 220,000 lb. Ans. See Art. **10.**

(5) Annealing makes the chain tougher, more ductile, and less liable to break from sudden jerks. See Art. **16.**

(6) Untwist a portion of the rope and examine the condition of the inner surfaces. See Art. **20.**

(7) (*a*) The safe load to be lifted depends on the size of the rope and on its manner of attachment to the hook of the tackle block or to the work. See Art. **23.**

(*b*) See Art. **23.**

(8) (*a*) Manila rope is chiefly used for hoisting and for power transmission. See Art. **26.**

(*b*) The fibers are oiled to reduce internal wear. See Art. **26.**

(9) The diameter of the pulley should be at least 40 times the diameter of the rope. See Art. **28.**

(10) Applying rule 4, Art. **30,**

$$W = 600 \times (1\tfrac{1}{2})^2 = 1{,}350 \text{ lb.} \quad \text{Ans.}$$

(11) Applying rule 6, Art. **30,**

$$C = \sqrt{16{,}000} \times .0316 = 4 \text{ in., nearly.} \quad \text{Ans.}$$

(12) A column both of whose ends are fixed is 4 times as strong as one that has both ends movable and $1\frac{7}{9}$ times as strong as one that has one fixed end and one movable end. See Art. **34.**

(13) See Art. **34.**

(14) Applying rule 7, Art. **37,** and taking the proper values from Tables V and VII, we have

$$W = \frac{14{,}000 \times .7854 \times 14^2}{1 + \dfrac{(16 \times 12)^2}{281.25 \times 14^2}} = 1{,}291{,}480 \text{ lb.} \quad \text{Ans.}$$

(15) By rule 8, Art. **42,** the area of the piston rod should be

$$\frac{.7854 \times 40^2 \times 110}{3{,}600} = 38.397 \text{ square inches.}$$

The corresponding diameter is $\sqrt{\dfrac{38.397}{.7854}} = 7$ inches, nearly. Hence a 6″ piston rod is too small. Ans.

(16) From Table X, S for this case is found to be 1,460. From Table XI, $R = \frac{14 \times 14^2}{6} = 457.33$. Now, applying the proper formula from Table IX, we have

$$W = \frac{4 \times 1{,}460 \times 457.33}{12 \times 12} = 18{,}547 \text{ lb.} \quad \text{Ans.}$$

(17) By rule **9,** Art. **50,** and Table XII, the safe load would be $4 \times 6 \times 500 = 12{,}000$ pounds. Hence, a load of 10,000 pounds is safe. Ans.

(18) From Table XII, $S = 4{,}400$. Applying rule **10,** Art. **51,** we have

$$a = \frac{400{,}000}{4{,}400} = 91 \text{ sq. in., nearly.} \quad \text{Ans.}$$

(19) Double shear means that the body subjected to the shearing stress resists this stress at two planes, at both of which shear must occur to produce failure. See Art. **48.**

(20) Countershafts serve to effect changes in speed and to stop and start the machinery. See Art. **54.**

(21) Cold rolled shafting is made cylindrically true by a special rolling process. Bright shafting is turned up in a lathe. See Art. **55.**

(22) The diameter by which bright shafting is designated is that of the bar from which it is turned. See Art. **56.**

(23) Pulleys should be placed as near the bearings as possible so that the deflection of the shaft may be as small as possible. See Art. **58.**

(24) Applying rule **11,** Art. **60,** after finding from Table XIV that C for this case is 85, we have

$$H = \frac{8^3 \times 80}{85} = 481.8, \text{ nearly, say } 482 \text{ H. P.} \quad \text{Ans.}$$

(25) From Table XIV we find C for this case to be 65. Applying rule **12,** Art. **61,**

$$R = \frac{65 \times 75}{4^3} = 76 \text{ rev.} \quad \text{Ans.}$$

(26) From Table XIV, $C = 95$. Applying rule **13**, Art. **62**,

$$D = \sqrt[3]{\frac{95 \times 84}{300}} = \sqrt[3]{26.6}, \text{ say 3 in. Ans.}$$

(27) The horsepower of a shaft will vary directly with the speed. See Art. **63.**

(28) See Art. **11.**

(29) A steel rope wears better than an iron rope. See Art. **29.**

HEAT AND STEAM.

(1) It causes a rise in temperature, causes an increase in volume, and causes a body to change from a solid to a liquid or from a liquid to a gas, according to the state it is in when the heat is applied. See Art. **8.**

(2) When we speak of the temperature of a body, we refer to the condition of the molecules that compose it. If they are in a state of rapid vibration, the body is said to have a high temperature; if they are moving slowly, it is said to have a low temperature. See Art. **2.**

(3) (*a*) A British thermal unit is the quantity of heat required to raise 1 pound of water through 1 degree Fahrenheit. See Art. **10.**

(*b*) One B. T. U. is equivalent to 778 foot-pounds of work. See Art. **13.**

(4) If two blocks of wood are rubbed briskly together, they will develop heat. There are many other similar instances. See Art. **12.**

(5) (*a*) and (*b*) See Art. **16.**

(*c*) See Art. **14.**

(6) To raise 10 pounds of water from 74° to 75° requires 10 B. T. U. Then, the specific heat of glass, according to Art. **15,** is the ratio of 1.77 to 10, or .177 equals the specific heat of glass. Ans. See Art. **15.**

(7) Since it takes 1 B. T. U. to raise 1 pound of water through 1 degree of temperature, and when 1 pound of water falls 1 degree of temperature, it gives off 1 B. T. U., 1,000 pounds of water falling 1 degree of temperature will give off 1,000 B. T. U., and falling from 212° to 192°, or 20°, will give off 1,000 × 20 = 20,000 B. T. U. Each pound of steam in being transformed into water gives off its latent heat, 966 B. T. U.; 1,000 pounds give off 1,000 × 966 = 966,000 B. T. U.; so the whole amount of heat liberated by 1,000 pounds of steam at 212° in being transformed to water at 192° is 966,000 + 20,000 = 986,000 B. T. U. In order to raise 1 pound of water from 32° to 112°, it takes 112° − 32° = 80 B. T. U. Therefore, 986,000 B. T. U. will raise 986,000 ÷ 80 = 12,325 lb. of water from 32° to 112°. Ans. See Arts. **9**, **10**, and **17**.

(8) Applying rule 2, Art. **19**, the product of the weight, specific heat, and temperature of the water is 10 × 1 × 90 = 900; of the mercury, 15 × .0333 × 60 = 29.97; of the alcohol, 20 × .60 × 40 = 480; and the sum of these products is 900 + 29.97 + 480 = 1,409.97. The product of the weight and specific heat of the water is 10 × 1 = 10; of the mercury, 15 × .0333 = .4995; of the alcohol, 20 × .60 = 12; and the sum of these products is 10 + .4995 + 12 = 22.4995. Then the temperature of the mixture is 1,409.97 ÷ 22.4995 = 62.66°. See Art. **19**.

(9) The quantity of heat contained in a mixture is the same as the quantity of heat contained in the substances before mixing. See Art. **19**.

(10) Using rule 3, Art. **20**, and substituting, we have

$$T = \frac{20\,(966 + 212) + 400 \times 55}{20 + 400} = 108.47^\circ. \quad \text{Ans.}$$

(11) The boiling point of a liquid varies with the pressure on its surface. See Art. **22**.

(12) (*a*) Saturated steam is steam having the temperature corresponding to its pressure. See Art. **23**.

(*b*) Superheated steam is steam that is not in contact with water and is at a temperature higher than the boiling point corresponding to its pressure. See Art. **25.**

(13) See Art. **27.**

(14) According to rule 1, Art. **15,** the heat required to raise the temperature of 10 pounds of ice from 25° to 32° is $.504 \times 10\,(32 - 25) = 35.28$ B. T. U. It requires 144 B. T. U. to change 1 pound of ice to water at the same temperature. To change 10 pounds of ice at 32° to water at 32° requires, therefore, $10 \times 144 = 1{,}440$ B. T. U. Then, to raise the water from 32° to 212° requires, according to rule 1, Art. **15,** $1 \times 10\,(212 - 32) = 1{,}800$ B. T. U. To change 1 pound of water into steam at the same temperature requires 966 B. T. U., and to change 10 pounds of water at 212° to steam at 212° requires $10 \times 966 = 9{,}660$ B. T. U. Therefore, to change 10 pounds of ice at 25° to steam at 212° requires

$$\begin{array}{r} 35.28 \\ 1440.00 \\ 1800.00 \\ 9660.00 \\ \hline 12935.28 \end{array} \text{ B. T. U. Ans.}$$

(15) See Art. **29.**

(16) 90 pounds gauge pressure $= 90 + 14.7 = 104.7$ pounds absolute pressure. The heat required to change water at the boiling temperature into steam at the same temperature, called the "latent heat of vaporization," is denoted in the table by L. Now, referring to the table, we find that for $p = 104.7$, $L = 881.368$. That is, it takes 881.368 B. T. U. to change 1 pound of water at 90 pounds gauge pressure into steam at the same pressure. To change 40 pounds requires $40 \times 881.368 = 35{,}254.72$ B. T. U. $35{,}254.72 \times 778 = 27{,}428{,}172$ ft.-lb. Ans.

THE STEAM ENGINE.

(1) The head end of a cylinder is the end farthest away from the crank-shaft; the crank end is the end nearest the crank-shaft. See Art. **4.**

(2) The distance passed over by the piston in moving from one extreme position in the cylinder to the other. See Art. **4.**

(3) The cylinder with its heads and steam chest, the steam pipe and exhaust pipe, the guides, the bed, and the bearings. See Art. **6.**

(4) The mechanism by which the steam is distributed, when considered as a whole, is termed the valve gear. See Art. **6.**

(5) The linear distance between the piston and the cylinder head when the former is at the end of its stroke is the piston clearance. The volume of this space added to the volume of the steam port leading to it is the clearance volume. See Art. **7.**

(6) The diameter of the circle described by the center of the eccentric is generally defined as the throw, but some people consider the eccentricity as the throw. See Art. **12.**

(7) By outside lap is meant the amount that the outside edge of the valve projects beyond the edge of the steam port when in mid-position; by inside lap is meant the amount that the inside edge of the valve projects beyond the edge of the steam port when in mid-position. See Art. **15.**

(8) No. See Art. **16.**

(9) The angle the eccentric radius makes with the position it would occupy if the valve had neither outside lap nor lead. See Art. **20.**

(10) Increase the outside lap of the valve, increase the valve travel, and the angle of advance. See Arts. **29, 60, 62,** and **63.**

(11) The exhaust port will be opened later and closed earlier. See Art. **30.**

(12) Lead is the distance the steam port is opened when the piston is at the end of its stroke. See Art. **31.**

(13) Ahead, irrespective of whether the engine runs under or over. See Art. **32.**

(14) By using a rocker-arm. See Art. **34.**

(15) No. The eccentric must be placed behind the crank. See Art. **35.**

(16) The exhaust port will be closed later on the forward stroke than on the return stroke, and hence there will be more compression at the head end. See Art. **41.**

(17) To double the port opening. See Arts. **42** and **43.**

(18) See Art. **46.**

(19) The volume swept through by the piston is $14^2 \times .7854 \times 28 = 4{,}310$ cubic inches. Then, the clearance is $\frac{247 \times 100}{4{,}310} = 5.73$ per cent. Ans. See Art. **51.**

(20) The apparent cut-off is $\frac{21 \times 100}{60} = 35$ per cent. Applying rule **1,** Art. **55,** we find the real cut-off to be $\frac{(35 + 2) \times 100}{100 + 2} = 36.27$ per cent. Ans.

(21) The ratio of expansion, by Art. **56,** is $\frac{100}{36.27} = 2.76.$ Ans.

(22) Since the lap circle is smaller, without having changed the position of its center, the point *d* of Fig. 25 of

the text will be farther to the right, and, consequently, the perpendicular dropped from it will also be farther to the right, thus showing that cut-off will take place later. Since the circumference of the lap circle is at a greater distance than before from the line $a\,b$, the lead line $g\,h$ is also farther away from it, which shows that the lead will be greater. The points of release and exhaust closure will not be affected at all, but the port opening will be greater. In case the port was opened originally its full width, the port opening obviously cannot be greater now than the width of the port; in this case it shows that the valve overtravels, i. e., moves in the same direction in which it moved to open the port after the port is wide open. The amount of overtravel will be equal to the difference in the width of the port and the distance of the circumference of the lap circle from the center o. See Arts. **60, 61,** and **62.**

(23) The piston speed is $\frac{80}{12} \times 75 \times 2 = 1{,}000$ feet per minute. The area of the piston is $44^2 \times .7854 = 1{,}520$ square inches, nearly. Applying rule **2,** Art. **66,** we get

$$c = \frac{1{,}520 \times 1{,}000}{6{,}000} = 253.3 \text{ square inches}$$

as the area of the steam pipe. The corresponding diameter is $\sqrt{\frac{253.3}{.7854}} = 18$ inches, nearly. By applying rule **3,** Art. **67,** we get

$$b = \frac{1{,}520 \times 1{,}000}{4{,}000} = 380 \text{ square inches}$$

as the area of the exhaust pipe. The corresponding diameter is $\sqrt{\frac{380}{.7854}} = 22$ in., nearly. Ans.

(24) The piston speed is 1,000 feet per minute. Applying rule **4,** Art. **68,** we get

$$a = \frac{1{,}520 \times 1{,}000}{7{,}500} = 202.7 \text{ sq. in.} \quad \text{Ans.}$$

THE INDICATOR.

(1) The pressure per square inch required to raise the pencil 1 inch on the paper drum. See Art. **3.**

(2) About equal to one-half the boiler pressure. See Art. **3.**

(3) A reducing motion is a mechanism used to communicate the motion of the crosshead, on a reduced scale, to the paper drum of an indicator. See Art. **10.**

(4) Their joints are apt to become loose and thus distort the diagram. See Art. **21.**

(5) By Art. **19,** $UV = \frac{40 \times 3}{24} = 5$ in. Ans.

(6) See that it is clean and in working order. The joints of the various links should be free but not slack enough to allow the pencil to shake. Also, see that there is no backlash between the piston and spring. See Art. **27.**

(7) The vacuum line is the line of no pressure and is located below and parallel to the atmospheric line at a distance equal to the atmospheric pressure divided by the scale of the spring used to make the diagram. See Art. **31.**

(8) Corliss diagrams have the events of the stroke sharply defined and the expansion and compression curves are comparatively even. Diagrams from high-speed engines do not show the events so distinctly and the expansion and compression lines are usually irregular. See Art. **34.**

(9) (*a*) Decrease the angular advance of the eccentric. See Art. **36.**

(*b*) Cut-off, release, and compression will be later. See Art. **36.**

(10) Change the length of the valve stem so as to make the cut-off occur earlier on the end doing the greater amount of work. See Art. **44.**

(11) (*a*) The higher the speed, the more compression is needed. See Art. **49.**

(*b*) $\frac{9}{10}$ the initial pressure with high-speed engines, $\frac{5}{10}$ with medium-speed engines, and from $\frac{2}{10}$ to $\frac{3}{10}$ with slow-speed engines. See Art. **49.**

(12) (*a*) There is probably some restriction in the steam passage leading from the boiler to the cylinder. See Art. **56.**

(*b*) The free escape of the exhaust is prevented. See Art. **57.**

(*c*) The valve leaks and allows steam to enter after cut-off. See Art. **53.**

(13) (*a*) Admission, release, and compression are all too late. The back pressure is also excessive.

(*b*) Shift the eccentric ahead on the shaft and make the exhaust port or exhaust pipe larger.

(14) The rounding of the corners at the beginning of the stroke indicates that admission is a trifle late, otherwise the diagrams are very good. See Art. **37.**

(15) On account of the reevaporation of some of the condensed steam near the end of the stroke. See Art. **52.**

(16) By prolonging the expansion line and noting where it leaves the actual line of the diagram. See Art. **54.**

(17) (*a*) To the vibrations of the pencil motion when there is a sudden change of pressure. See Art. **58.**

(*b*) To the sticking of the indicator piston. See Art. **59.**

(18) Compression is too early on the head end, as is shown by the compression line extending above the steam line. There is also an unequal distribution of power in the two ends of the cylinder, as shown by the cut-off being much later on the crank-end diagram, which shows that the valve gear has been displaced. Release is too late on the crank-end diagram. See Arts. **42** and **44.**

(19) The valve on the engine from which Fig. 20 was taken had no lead, while that from which Fig. 22 was taken had lead. See Art. **42.**

(20) No. The steam might leak out as fast as it leaks in. See Art. **53.**

(21) Admission, cut-off, and release are too late on the crank end of the cylinder, thus giving an excess of power above the head end.

ENGINE TESTING.

(1) (*a*) When a gas expands and does work and no heat is added to it from an outside source the expansion is said to be adiabatic. See Art. **7.**

(*b*) When the temperature of a gas is kept constant during expansion, the expansion is said to be isothermal. See Art. **8.**

(2) The work is the same. See Art. **11.**

(3) If dry, saturated steam expands adiabatically, some of it will be condensed, and the relation between pressure and volume will then depend on the proportion of water present in the mixture. If, on the other hand, the expansion is isothermal, the steam will be superheated. See Art. **17.**

(4) (*a*) By rule **1**, Art. **20**, the work is

$$W = 144\,PV = 144 \times 45.6 \times 6.3 = 41,368.32 \text{ ft.-lb.} \quad \text{Ans.}$$

(*b*) The number of foot-pounds per minute is 41,368.32 × 76, and the horsepower developed is $\frac{41,368.32 \times 76}{33,000} = 95.27$ H. P. Ans.

(5) (*a*) The mean ordinate is the ordinate whose length is the average of all the ordinates of the diagram. See Art. **24.**

(*b*) It is found by dividing the area of the diagram by its length. In case the area is not known, divide the length of the diagram into a number of equal parts, and half way between these parts or points of division draw vertical lines extending from the upper to the lower lines of the diagram. To find the mean ordinate, add the lengths of these ordinates or vertical lines and divide by their number. See Arts. **24** and **26.** It may also be found by a planimeter.

(6) (*a*) Volume of cylinder = area × length = 1 × 2 = 2 cubic feet. The horizontal scale of volumes

$$= \frac{\text{volume of cylinder in cubic feet}}{\text{length of card in inches}} = \tfrac{2}{3} \text{ cu. ft. per in.} \quad \text{Ans.}$$

See Art. **21.**

(*b*) Work per stroke of piston = area × horizontal scale × vertical scale × 144 or work = $6.75 \times \frac{2}{3} \times 50 \times 144 = 32{,}400$ ft.-lb. Ans. See Art. **26.**

(7) By taking the difference between the indicated horsepower when the engine is running loaded and the indicated horsepower when it is running light. See Art. **32.**

(8) The approximate net H. P. = I. H. P. − friction H. P. = 176.8 − 25.6 = 151.2. The efficiency by rule **2**

$$= \frac{151.2}{176.8} = .855 = 85.5 \text{ per cent.} \quad \textbf{Ans.}$$

See Art. **33.**

(9) $\frac{2.75}{3.15} \times 50 = 43.65$ lb. per sq. in., M. E. P. Ans.

See Art. **36.**

(10) By passing around the diagram two or three times and noting the reading at the return to the starting point, each time. The difference between the readings should be the same each time or the last reading divided by the number of readings should equal the first reading. See Art. **41.**

(11) By subtracting the sum of the lengths of the ordinates of the loops from the sum of the lengths of the ordinates of the main part of the diagram and dividing by the number of ordinates. See Art. **45.**

(12) Using rule **3** and the table, Art. **46,** 75 + 14.7 = 89.7. From the table the constant for $\frac{5}{8}$ cut-off is .927, and

.927 × absolute pressure = .927 × 89.7 = 83.15. M. E. P. = (83.15 − 17) × .9 = 59.54 lb. per sq. in. Ans.

(13) (*a*) Piston speed is the distance traveled by the piston in 1 minute.

(*b*) By rule **5,** Art. **48,** $S = \frac{24 \times 180}{6} = 720$ ft. per min. Ans.

(14) It is a number obtained by combining into a single factor all the constant horsepower factors for that engine. See Art. **50.**

(15) (*a*) The length L of the stroke is $\frac{24}{12} = 2$ feet; the area A of the piston is $18^2 \times .7854 = 254.47$ square inches, and the number of strokes N is $2 \times 185 = 370$. Substituting these values in the formula corresponding to rule **6,** we have

$$C_u = \frac{\frac{24}{12} \times 254.47 \times 185 \times 2}{33,000} = 5.706. \quad \text{Ans.}$$

(*b*) Multiplying the engine constant by the M. E. P., we have I. H. P. = 5.706 × 53.8 = 306.98. Ans. See Art. **51.**

(16) (*a*) Scale of spring $s = 60$; length L of stroke is $\frac{36}{12} = 3$ feet; area A of piston $= 24^2 \times .7854 = 452.39$ square inches, and the number N of working strokes is $2 \times 150 = 300$ per minute. The number n of ordinates is 20. Substituting these values in rule **10,** we have

$$C_o = \frac{60 \times 3 \times 452.39 \times 300}{33,000 \times 20} = 37.013. \quad \text{Ans.}$$

(*b*) The I. H. P. = 37.013 × 16 = 592.2. Ans. See Art. **56.**

(17) The horsepower measured by some type of absorption dynamometer. See Art. **60.**

(18) The net pressure on the scale = 274.5 − 14.5 = 260 pounds. Substituting, in rule **11,** Art. **61,** we have

$$\text{H. P.} = \frac{260 \times 4 \times 225 \times 6.2832}{33,000} = 44.55. \quad \text{Ans.}$$

(19) The lever arm in this case is $\frac{\frac{47}{2} + \frac{1}{2}}{12} = 2$ feet and the net pull is $241 - 10 = 231$ pounds. Substituting in rule **11,** Art. **61,** we have

$$\text{H. P.} = \frac{231 \times 2 \times 350 \times 6.2832}{33{,}000} = 30.78. \quad \text{Ans.}$$

(20) When fresh steam enters the cylinder, part of it is condensed and is not taken account of by the indicator diagram. See Art. **65.**

(21) By calculating the water consumption at cut-off and then at release. See Art. **67.**

(22) Length of stroke $= \frac{20}{12} = 1\frac{2}{3}$ feet; hence, each inch of length of diagram equals $\frac{1\frac{2}{3}}{3.25} = .51$ foot of stroke. As a 45-pound spring was used to make the diagram, $a\,m = .70 \times 45 = 31.5$ pounds and $b\,n = .62 \times 45 = 27.9$ pounds. Also, $O\,m = 3.41 \times .51 = 1.74$ feet and $O\,n = .36 \times .51 = .18$ foot. Area of piston $= \frac{16^2 \times .7854}{144} = 1.396$ square feet. The volume of steam in the cylinder when the piston is at the point represented by a is $1.74 \times 1.396 = 2.429$ cubic feet. The volume when the piston is at b is $.18 \times 1.396 = .25128$ cubic foot. The weight of a cubic foot of steam at an absolute pressure of 31.5 pounds is .07768 pound, and at a pressure of 27.9 pounds the weight is .06931 pound. The weight of steam in the cylinder is $.07768 \times 2.429 = .18868$ pound and the weight saved by compression is $.06931 \times .25128 = .017416$ pound. The steam used per stroke is $.18868 - .017416 = .17127$ pound. See Art. **66.**

The M. E. P. $= \frac{2.41 \times 45}{3.25} = 33.37$ pounds. See Art. **36.**

The I. H. P. is $\frac{33.37 \times 1\frac{2}{3} \times 16^2 \times .7854 \times 160 \times 2}{33{,}000} = 108.4.$ See Art. **47,** rule **4.** The water consumption per I. H. P. per hour is $\frac{.17127 \times 160 \times 2 \times 60}{108.4} = 30.34$ lb. See Art. **66.**

(23) The pressure at *a* is $50 \times .71 = 35.5$ pounds, absolute. The weight of a cubic foot of steam at this pressure is .086916 pound. Using rule **12,** we have

$$Q = \frac{13,750 \times 2.93 \times .086916}{37.5 \times 3.36} = 27.79 \text{ lb. per I. H. P. per hr.}$$

See Art. **68.** Ans.

GOVERNORS.

(1) When the turning effect on the crank is greater than the average resistance, part of the energy is absorbed in increasing the speed of the flywheel; but when the turning effect is less than the average resistance, the flywheel gives up part of its energy and thus tends to keep the speed constant until the governor has time to act. See Art. **2.**

(2) The pressure of the steam admitted to the cylinder is varied or the point of cut-off is changed. See Art. **3.**

(3) When the product of the centrifugal force multiplied by the distance h is equal to the product of the weight of the ball multiplied by the distance r. See Art. **7.**

(4) At high speeds the change in height is so small that the controlling valve cannot be moved through a great enough distance. See Art. **10.**

(5) When the change of height of a governor is increased for a given change of speed, its sensitiveness is said to be increased. See Art. **12.**

(6) (*a*) The small forces acting on the weight of a weighted-pendulum governor require some time to overcome the inertia of the weight. With the spring-loaded governor, the inertia of the parts to be overcome by the centrifugal force of the balls is very small; consequently, it acts more quickly than the weighted-pendulum governor. See Art. **15.**

(*b*) The inertia of the parts of the spring-loaded governor being very small as compared to the force moving them, the governor is not so apt to be carried beyond its proper position for each change in speed as is the weighted-pendulum governor, whose moving forces are weak and whose parts are heavy. See Art. **15.**

(7) The maximum port opening is equal to the lead. See Art. **17.**

(8) (*a*) As the eccentric swings towards the center of the shaft the lead is increased.

(*b*) It is an advantage to have an increase in the lead as the cut-off becomes earlier, to prevent wiredrawing. This method of suspension has the disadvantage of tending to permit the engine to run away when the load is suddenly thrown off. See Art. **25.**

(9) By locating the pivot on the same side of the shaft as the crank and to one side of the center line of the crank. See Art. **26.**

(10) The inertia of the weights and bar tends to keep them moving at a constant velocity, so that if the speed of the engine is suddenly increased or decreased, the governor case will move ahead or drop behind the bar, as the case may be. This movement, which is equivalent to a motion of the bar about its pivot, adjusts the cut-off. See Art. **31.**

(11) By connecting the eccentric to an arm separate from the inertia bar and then connecting this arm to the inertia bar in such a manner that its direction of rotation, with respect to the governor case, will be opposite to that of the inertia bar. See Art. **32.**

(12) There is but one speed at which the weights of a perfectly isochronous governor will be in equilibrium, and at that particular speed the weights will remain in any position of their travel; consequently, a slight change in speed will cause them to move to one or the other extreme position. See Art. **34.**

(13) By adjusting the tension on the spring so that its resistance will increase somewhat faster than the centrifugal force. See Art. **36.**

(14) The weights of the governor cannot move until their speed has been increased sufficiently to overcome the frictional resistance, so that when the weights do move they are likely to move too far, as the frictional resistance is generally reduced when the parts are in motion. See Art. **37.**

(15) Applying rule **1,** Art. **40,** we have

$$D_2 = \frac{5 \times 180}{162} = 5.55 \text{ in.} = 5\tfrac{9}{16} \text{ in., nearly.} \quad \text{Ans.}$$

(16) Any change in the tension on the springs has an effect on the sensitiveness of the governor; consequently, if the speed of the engine is to be changed greatly, the governor is liable to become too sensitive or not sensitive enough. See Art. **42.**

(17) No. The change in centrifugal force causing the weights to fly outwards will be neutralized by the change in the force of gravity tending to draw the weights downwards, so that the weights will remain in the same position, no matter how much their mass is changed. See Arts. **7** and **39.**

(18) The weights must be made lighter. See rule **4,** Art. **50.**

(19) A self-locking governor is one in which the position of the weights is locked so that the force acting on the eccentric cannot move them. See Art. **56.**

(20) The travel of the valve is reduced and the angle of advance is increased. See Art. **18.**

(21) The ratio between the centrifugal force of the weights and the opposing resistance must be constant for all positions of the weights. See Art. **33.**

(22) By proportioning the springs so that their tension will increase slower than the centrifugal force of the weights. See Art. **46.**

VALVE GEARS.

(PART 1.)

(1) Slide valves, rotary valves, and poppet valves. See Art. **8.**

(2) To give a greater range of cut-off without affecting the points of admission, release, and compression. See Art. **9.**

(3) The practical range of cut-off is very small, and the space in the main-valve chamber is filled with steam at cut-off which destroys some of the benefits of early cut-off by increasing the clearance during the period of expansion. See Art. **11.**

(4) The eccentric controlling the main valve is set 90° + the angle of advance ahead of the crank, the same as for a plain slide valve. The auxiliary eccentric is set nearly opposite the crank. See Art. **13.**

(5) After setting the main valve so that the lead is the same at both ends, turn the engine to the point at which it is desired to have cut-off take place; then turn the cut-off eccentric, until the edge of the cut-off valve just closes the passage of the main valve that is open to the cylinder; then fasten the eccentric. See Art. **15.**

(6) The plain **D** slide valve requires considerable force to drive it, which not only absorbs considerable of the power of the engine, but greatly affects the action of the governor. When used with a governor which changes the throw of the eccentric, the port opening is greatly restricted at early

cut-offs, which produces wiredrawing. If the valve is given overtravel to overcome this latter difficulty, the valve has to be made larger and requires still more power to move it.

(7) In horizontal engines the wear due to the weight of the valve does not affect the tightness of the valve, and if the valve does wear so that it leaks, it can usually be made tight without renewing any of the parts. The pressure-plate valve also can be made so as to act as a relief valve for the escape of water from the cylinder. See Arts. **20** and **21.**

(8) The steam should be admitted to the steam chest slowly so as to warm the valve and its seat gradually and thus prevent the valve from binding in its seat. See Art. **21.**

(9) Not very seriously. The space back of the valve is connected with the exhaust port so that the pressure in this space is always the same as the pressure in the exhaust port. See Art. **22.**

(10) To quickly close the steam valve after it has been disconnected from the cut-off mechanism. See Art. **29.**

(11) The steam valves are so arranged that, when they are moving at their most rapid rate, steam is being admitted to the cylinder, their period of slow motion corresponding to the time during which they are closed. In a similar manner the most rapid motions of the exhaust valves take place at release and compression. See Arts. **31** and **32.**

(12) The valve will work as though positively connected to the admission crank. See Arts. **34** and **40.**

(13) Suppose the valves do not have lap or lead and the releasing gear does not work, then the steam valve closes at the extreme end of the stroke and the corresponding exhaust valve opens at the same instant, thus giving the steam a chance to blow through. This occurs on both ends of the stroke. It is necessary, then, to give the steam valve sufficient lap to close it before the exhaust valve opens and

prevents its opening until the exhaust valve has closed. In order to have prompt release and sufficient compression to check the motion of the reciprocating parts of the engine, the exhaust valves must be given a certain amount of lead. See Art. **40.**

(14) When a single eccentric is used, it is necessary to give it a considerable angle of advance in order to give a proper steam distribution. As the releasing mechanism cannot act later than ½ stroke, this angle of advance greatly reduces the effective range of cut-off. By the use of two eccentrics, one for the exhaust valves and one for the steam valves, the range through which the governor can control the cut-off is greatly extended, as the cut-off eccentric can be set 90°, or even a little less, ahead of the crank. See Art. **49.**

(15) The steam valves will open very slowly at the beginning of the stroke, thus producing wiredrawing of the steam. See Art. **50.**

(16) The so-called high-speed Corliss engine has the advantage of the four oscillating valves and wristplate motion of the ordinary Corliss engine, and as its valves are positively driven, they can operate at a much higher speed. The degree of regulation is also superior because the wristplate is driven by a shifting eccentric controlled by a shaft governor. By the use of two eccentrics all the other advantages of the ordinary Corliss gear, using two eccentrics, may be obtained. See Art. **53.**

(17) By changing the length of the carrier rod, so as to move the wristplate towards its center of motion, by an amount equal to ½ the overtravel. See Art. **43.**

(18) Cut-off is made earlier; the other events remain the same. See Art. **13.**

(19) Overtravel is the amount the valve travels over that which is necessary to give a full port opening. See Art. **18.**

(20) To secure an increased port opening with a small valve travel. See Art. **18.**

(21) Shorten the eccentric rod until the valve has the same lead on both ends; then decrease the angle of advance of the eccentric until the required lead is obtained. See Art. **15.**

STEAM TURBINES

(1) Yes. See Arts. **3** and **9.**

(2) To close the opening in the casing through which the shaft passes. See Art. **5.**

(3) To confine the shaft longitudinally. See Art. **5.**

(4) A vacuum valve is fitted, which admits air to the casing, the air acting as a brake upon the turbine wheel. See Art. **7.**

(5) It permits the turbine wheel to run at a very high speed without vibration. See Art. **8.**

(6) It greatly increases the velocity of the steam discharged from it. See Art. **9.**

(7) Yes. In the De Laval turbine expansion takes place before the steam impinges on the blades ; in the Westinghouse-Parsons turbine expansion takes place while the steam passes from one set of blades to the other. See Art. **11.**

(8) By careful balancing and the use of bearings capable of lateral accommodation. See Art. **14.**

(9) No. Steam is admitted in puffs. See Art. **16.**

(10) It is first expanded in expansion nozzles, and then in passing from one set of blades to the next. See Arts. **17** and **18.**

(11) Only the last set. See Art. **19.**

(12) In the steam turbine the steam gives up its kinetic energy and thus does work ; in the reciprocating engine the pressure of the steam is transformed into work. See Art. **20.**

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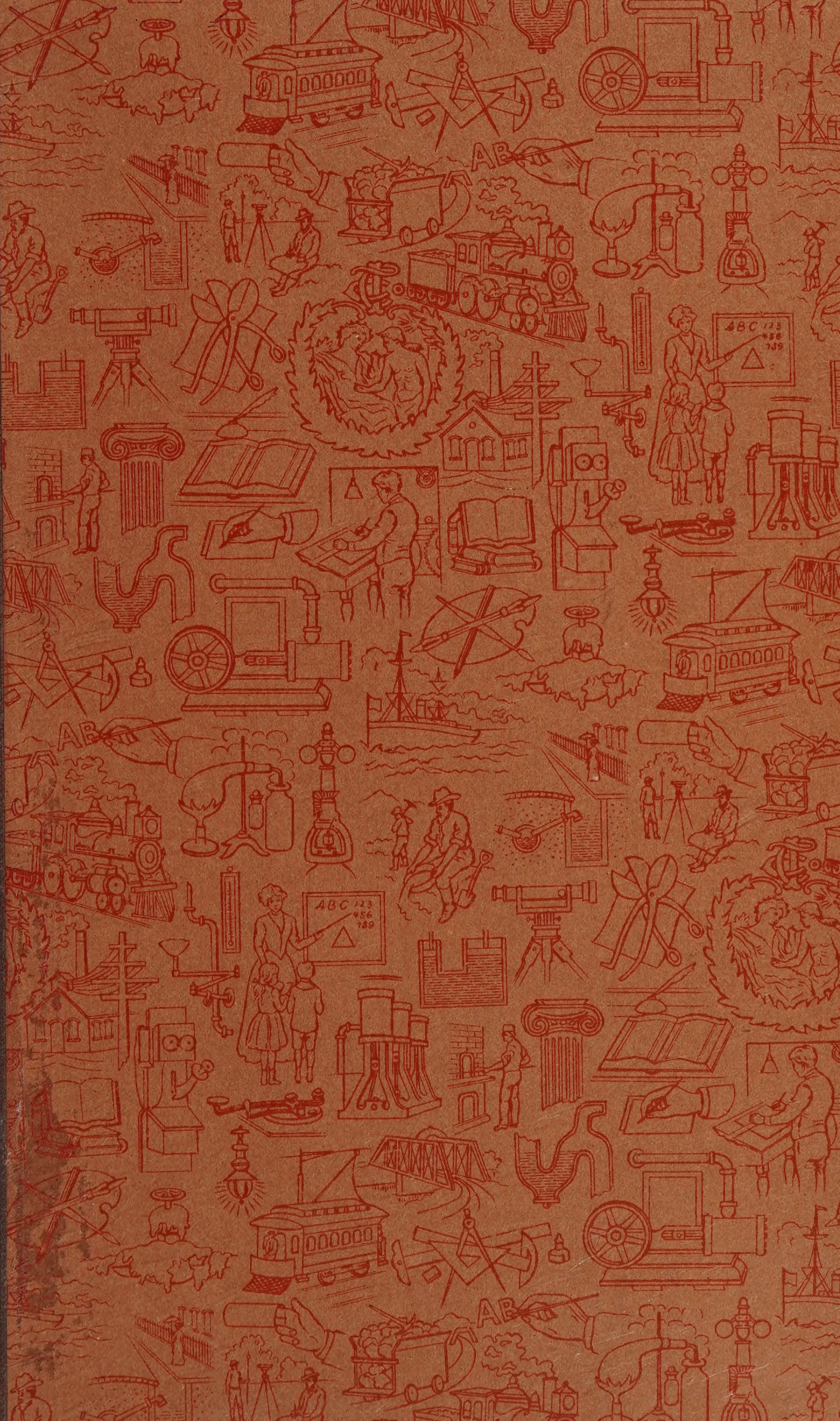
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